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In-service testing of heavy vehicle suspensions – background report for the NTC project

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Executive Summary

Some road authorities are required, by agreement with the Australian Government, to develop an in-service HV suspension test (Australia Department of Transport and Regional Services, 2005a, 2005b). Development and implementation of such in-service HV suspension testing procedures, who would perform them and field test equipment will be the outcomes of a project currently underway and managed by the National Transport Commission (NTC). This report is a background briefing on suspension testing techniques for the NTC project.

The “science” of pavement and bridge behaviour is not exact. Even so, models are used to estimate heavy vehicle (HV) wheel-force damage to the road network asset. Likewise, the “science” of HV suspension performance is not exact. These shortcomings did not stop “road-friendly” air-suspended HVs being allowed greater mass as a concession in Australia and Europe. This move, whilst questionable with the clarity of hindsight, now needs to be managed in the light of better understanding of the dynamics of HV air suspensions. This management approach needs to consider that the test regime for new HV suspensions in Australia could benefit from revision, dating as it does from before HV air suspension behaviour was better understood.

The role of suspension dampers (shock absorbers) is critical to the continuing health of HV air suspensions. Over 50% of air suspensions on HVs on the road do not meet the Australian requirement for “road-friendliness”. An in-service test is to be developed for air-suspended HVs, as agreed between two Australian States and the Commonwealth, to ensure HV damper health. By applying this, or other mechanisms, to ensure continued HV suspension health, the saving to Queensland Main Roads alone in terms of reduced maintenance costs will be greater than \$59M/annum (in 2007 \$). This quantum does not include the road safety or workplace health & safety impacts of out-of-specification HV suspensions. Neither does it include Local Government impacts.

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Definitions, Abbreviations & Glossary

Terms, abbreviations and acronyms	Meaning
ABDC	Austroads bridge design code.
Aggregate Force	A method for determining HV wheel-forces at a particular point on a length of pavement. For a length of instrumented pavement, the aggregate force, F at sensor k may be derived using instantaneous wheel forces on that pavement: $$F_k = \sum_{j=1}^{N_a} P_{jk} \qquad \text{for } k = 1, 2, 3, \ldots n$$ Where: F_k = instantaneous wheel-force at sensor k; N_a = number of axles on the vehicle; and P_{jk} = the force applied by wheel or tyre j to sensor k (Cebon, 1987; Cole, Collop, Potter, & Cebon, 1996).
ARRB	Australian Road Research Board – now privatised, has changed its name to ARRB Group Limited.
ARTSA	Australian Road Transport Suppliers Association.
ATC	Australian Transport Council. "The Australian Transport Council (ATC) is a Ministerial forum for Commonwealth, State and Territory consultations and provides advice to governments on the coordination and integration of all transport and road policy issues at a national level." http://www.atcouncil.gov.au
ATRF	Australasian Transport Research Forum. A conference for presentation of papers and colloquia on matters of transport planning, policy and research.

Axle hop	Vertical displacement of the wheels (and axle), indicating dynamic behaviour of the axle and resulting in more or less tyre force onto the road. Usually manifests in the frequency range 10 – 15Hz.
BIFA	Bilateral Infrastructure Funding Agreement. Also known as the Auslink agreement. An agreement between individual States of Australia and the Commonwealth which "covers arrangements applying to funding made available by the Australian Government to Queensland under the first five-year AusLink investment programme (2004-05 to 2008-09) and any agreed subsequent changes to, and extensions of, the programme. It also covers agreed arrangements for infrastructure planning, identification of investment priorities, development and assessment of project proposals and evaluation of completed projects." (Australia Department of Transport and Regional Services, 2005b). Queensland's BIFA may be viewed at: http://www.auslink.gov.au/publications/policies/pdf/Queensland_bilateral.pdf
Body bounce	Movement of the sprung mass of a truck as measured between the axles and the chassis. Results in truck body dynamic forces being transmitted to the road via the axles & wheels. Usually manifests in the frequency range 1 – 4Hz.
CoG	Centre of gravity. The point at which a body's mass may be said be concentrated for purposes of determining forces on that body.

Correlation coefficient	For any two dynamic wheel load measurements, their correlation function can be used determine the measure of spatial repeatability. For highly correlated wheel-force histories measured along tyre-paths during testing the maxima and minima will tend to occur at similar points on the travelled path. This effect can be defined using a measure defined as the correlation coefficient: $$\rho \propto \frac{[f_n(t) - \overline{f_n(t)}][f_{n+1}(t) - \overline{f_{n+1}(t)}]}{\sigma_{fn}\sigma_{fn+1}}$$ Where: ρ = the correlation coefficient; $f_n(t)$ = the n^{th} wheel-force time history signal; $\overline{f_n(t)}$ = the mean of the n^{th} wheel-force time history signal; and σ_{fn} = the standard deviation of the n^{th} wheel-force time history signal. See also spatial repeatability (Kenis, Mrad, & El-Gindy, 1998).
Damping ratio	How much the shock absorbers reduce suspension bounce after the truck hits a bump. The damping ratio, zeta (ζ) is given as a value under 1 (e.g. 0.3) or a percentage (e.g. 30%).
Δ	Greek letter “delta” – denoting increment.
DIF Dynamic impact factor	DIF = $\frac{PDF}{F_{stat(axle)}}$ Where: PDF = peak instantaneous force measured during the test; and $F_{stat\ (axle)}$ is the static axle force (Woodrooffe & LeBlanc, 1987). See also PDWF.
DIVINE	Dynamic Interaction between heavy Vehicles and INfrastructurE.
DoTaRS	Department of Transport and Regional Services. An Australian Government department.

Dot operator	Denotes derivative with respect to time. Where (say) x is a time-dependant variable, dx/dt denotes the derivative or the rate of change with respect to time. This concept is sometimes denoted $\dot{x}$ as a shortened form. The derivative with respect to time of $\dot{x}$, that is d^2x/dt^2 or acceleration, is denoted $\ddot{x}$.
Dynamic load allowance (DLA)	The dynamic load allowance (DLA) nominates an increment in the static design forces to allow for dynamic and resonant loads in bridge design. It is influenced by the dynamic interactions between vehicles and structures. It similar to, but more complex in its derivation than, dynamic increment.
DLSC Dynamic load-sharing coefficient	The standard deviation of the function of instantaneous dynamic load-sharing: DLS*i*. The instantaneous forces at axle *i* are summed to get F_i for comparison with the other axle/s in a multi-axle group. DLSC = $\sqrt{\frac{\sum(DLS_i - \overline{DLS_i})^2}{k}}$ Where: Dynamic load-sharing (DLS) at axle *i*, $DLS_i = \frac{nF_i}{\sum_{i=1}^{i=n} F_i}$ n = number of axles; F_i = instantaneous wheel-force at axle i ;and k = number of instantaneous values of DLS, i.e. number of terms in the series (de Pont, 1997).

Dynamic Increment (DI)	Dynamic increment. The increment is the incremental value or fraction above the static load that is imparted to bridges by dynamic forces of vehicles. Various researchers such as Heywood defined the DI as: DI = $\frac{(\delta_{dyn} - \delta_{static}) \times 100\%}{\delta_{static}}$ Where: δ_{dyn} = peak dynamic deflection or strain in the structure; and δ_{static} = peak static deflection or strain in the structure (Cantieni, 1992; Heywood, 1995).
Dynamic load coefficient (DLC)	Coefficient of variation of dynamic tyre force. It is obtained by calculating the ratio of the root-mean-square (RMS) of the dynamic wheel forces (std. dev. of F_{mean}) divided by the static wheel-force, i.e. the coefficient of variation of the total wheel load: DLC = σ / F_{mean} Where: σ = the standard deviation of wheel-force; and F_{mean} = the mean wheel-force. A perfect suspension would have a DLC of 0. The range in reality is somewhere between 0 and 0.4 (Mitchell & Gyenes, 1989).
Eigenfrequency	Frequency of a body at one of its vibrational resonance modes.
FFT	Fast Fourier transform. A method whereby the Fourier transform is found using discretisation and conversion into a frequency spectrum.

Fourier transform	A method whereby the relative magnitudes of the frequency components of a time-series signal are converted to, and displayed as, a frequency series. If the integrable function is h(t), then the Fourier transform is: $$\phi(\omega) = \int_{-\infty}^{+\infty} h(t) e^{-i\omega t} dt$$ Where: ϕ is the Fourier series; ω is the frequency in radians/s; and $i = \sqrt{-1}$ (Jacob & Dolcemascolo, 1998).
GVM	Gross vehicle mass.
HML	Higher mass limits. Under the HML schemes in Australia, heavy vehicles are allowed to carry more mass (payload) in return for their suspension configuration being "road friendly". See VSB 11.
HV	Heavy vehicle.
Hz	Hertz. Unit of vibration denoting cycles per second.

LSC	Load-sharing coefficient – a measure of how well a suspension group equalises the total axle group load, averaged during a test. This is a value which shows how well the average forces of a multi-axle group are distributed over each tyre &/or wheel in that group. $$LSC = \frac{F_{mean}(i)}{F_{stat\ (nom)}}$$ Where: $F_{stat\ (nom)}$ = Nominal static tyre force = $\frac{F_{group\ \ (total)}}{n}$ $F_{group\ (total)}$ = Total axle group force; F_{mean} (i) = the mean force on tyre/wheel i ; and n = number of tyres in the group (Potter, Cebon, Cole, & Collop, 1996).
LVDT	Linear Variable Differential Transformer. A displacement transducer using a transformer with movable primary and secondary coils. Relative movement between the primary and secondary coils results in measurable variation in the level of alternating current excitation that is proportion to the amount of movement.
MCV	Multi-combination vehicle. HVs with general arrangement or GVM greater than that of a semi-trailer.
MLR	Mass Limits Review. The national project that resulted in the implementation of HML in Australia.
NHVAS	National Heavy Vehicle Accreditation Scheme. A voluntary scheme that certifies transport operators against a set of industry-specific quality assurance requirements. Membership of this scheme is a pre-requisite for HML.
NRTC	National Road Transport Commission. A national body set up by the States of Australia to facilitate economic reform of the road transport industry. Became the NTC earlier this decade.
NSW	New South Wales.

NTC	See NRTC
OECD	Organisation for Economic Co-operation and Development
PDLR	Peak Dynamic Load Ratio. PDLR = $\frac{PDWF}{F_{stat}}$ Where: $PDWF$ = peak dynamic wheel-force measured instantaneously during the test (see PDWF); and F_{stat} is the static wheel-force (Fletcher, Prem, & Heywood, 2002).
PDWF	Peak dynamic wheel-force. The maximum wheel-force experienced by a wheel during dynamic loading as a result of a step input (Fletcher *et al*., 2002). If applied to axle forces, this measure is related to dynamic impact factor (DIF) as the numerator in the equation.
QUT	Queensland University of Technology
RFS	"Road-friendly" suspension. A HV suspension conforming to certain limits of performance parameters defined by VSB 11. http://www.dotars.gov.au/roads/safety/bulletin/pdf/vsb_11.pdf
RSF	Road stress factor. An estimation of road damage due to the 4th power of instantaneous wheel-force given by: RSF = $(1+6DLC^2 + 3DLC^4)P^4_{stat}$ Where DLC is as above and P_{stat} is the static wheel-force (Potter, Cebon, & Cole, 1997).

σ	Greek symbol lower-case 'sigma', denoting standard deviation: $\sigma = \sqrt{\frac{\sum (x_i - \bar{x})^2}{n}}$ for $i = 1, 2, 3, \ldots n$ Where: x_i is a number in a series; $\bar{x}$ is the mean of the series; and n is the number of terms in the series.
Spatial repeatability	The tendency for HV suspensions with similar characteristics to concentrate dynamic wheel-forces at particular points on any given length of road. See also correlation coefficient.

SRI	Spatial repeatability index. The correlation coefficient of the aggregate tyre force histories of two signals. $$SRI = \frac{\frac{\omega}{2\pi V}\int_0^{\frac{2\pi V}{\omega}}[f(t)-\overline{f(t)}][g(t)-\overline{g(t)}]dx}{\sigma_f \sigma_g}$$ Where: SRI = spatial repeatability index; t = x/V; x = the distance along the measured wheel path/s; V = the velocity of the vehicle; f(t) = h(t) + ε_f(t) representing a reference wheel-force time history signal with frequency ω and an error signal ε_f; g(t) = k(t) + ε_g(t) representing test wheel-force time history signal with frequency ω and an error signal ε_g; h(t) = sin(ωt), a reference wheel-force time history signal as an assumed sinusoid; k(t) = sin(ωt + θ), a test wheel-force time history signal as an assumed sinusoid with phase-shift θ; σ_f = the standard deviation of the wheel-force time history signal f; σ_g = the standard deviation of the wheel-force time history signal g; $\overline{f(t)}$ = the mean of f(t); and $\overline{g(t)}$ = the mean of g(t) (Collop, Potter, Cebon, & Cole, 1994).

String potentiometer	A resistance device, the resistance of which varies depending on the linear extension of a string which is converted into angular displacement of its input shaft by the string being wound around a spring-loaded wheel on that shaft. The input shaft drives the wiper arm of a potentiometer. The angular displacement of the shaft is therefore proportional to the resistance of a potentiometer (variable resistor) and can be measured as linear extension of the string.
VSB 11	Vehicle Standards Bulletin 11. A document issued by DoTaRS that defines the performance parameters of "road-friendly" HV suspensions.
Weighted Aggregate Force	To find a measure of pavement damage the aggregate force F_k (see aggregate force) may be raised to the appropriate power q using any of the pavement damage "power rules". For a length of instrumented pavement, the aggregate force F at sensor k may be raised to the appropriate power to determine road damage due to instantaneous wheel forces on that pavement: $$F_k^{\ q} = \sum_{j=1}^{N_a} P^q{}_{jk} \qquad \text{for } k = 1, 2, 3, \ldots n$$ Where: F_k = instantaneous wheel-force at sensor k; N_a = number of axles on the vehicle; P_{jk} = the force applied by wheel or tyre j to sensor k; and q = the chosen exponent of road damage. This is termed the weighted aggregate force (Cebon, 1987).
WiM	Weigh-in-motion. Technology that uses sensors in the road to measure the wheel-force of vehicles.

1 Introduction

This report is a background briefing for the NTC heavy vehicle in-service suspension testing project. It has been condensed from a literature review for the project *Heavy vehicle suspensions – testing and analysis* currently underway at QUT.

1.1 Rationale

1.1.1 The need for HV "road-friendly" suspension testing

"Road-friendly" HV suspensions are critically dependant on correct shock absorber function. An in-service test for road friendliness would be an advantage to both the transport industry and road asset owners. The former because worn shock absorbers could be replaced before vehicle and payload damage occurs; high-mileage but still serviceable shock absorbers need not be replaced (saving labour and equipment costs). The latter because road and bridge asset rehabilitation costs would be reduced through less wear-and tear from HVs with nominally-"road-friendly" suspensions with out-of-specification or deficient shock absorbers. Development of such an in-service test has been mandated between two Australian States and the Commonwealth (Australia Department of Transport and Regional Services, 2005a, 2005b).

The anecdotal HV transport industry view of worn shock absorbers and the attendant issue of air suspension health is that the resultant tyre wear is detected quickly. This is then rectified to prevent further increased tyre wear and the associated costs of premature tyre replacement. Despite this view, the Marulan survey (Blanksby, George, Germanchev, Patrick, & Marsh, 2006) showed that more than half the HVs sampled on the Hume Highway did not meet at least one VSB 11 suspension parameter. This was confirmed in part by Sweatman *et al.*, (2000) who found that:

- quantitative evaluation of shock absorbers did not usually take place in most fleets; and
- the trigger for replacement of shock absorbers was visible leakage or lack of heat after a trip.

From this empirical evidence, it may be inferred that the industry indicators of using tyre wear, leakage or temperature to detect out-of-specification or deficient shock absorbers is too late in the maintenance cycle to be effective at meeting the Australian requirements for "road-friendly" HV suspensions. Compounding this issue is the fact that there are no recognised low-cost in-service HV suspension tests in Australia. This has been discussed previously (Starrs *et al.,* 2000, Sweatman *et al.,* 2000) without decisive action by regulators until recently. That action is now occurring on this issue is due to agreement between two Australian States and the Commonwealth (Australia Department of Transport and Regional Services, 2005a, 2005b).

1.2 Organisation of this report

Testing heavy vehicle dynamic measures includes testing suspension parameters and wheel forces. It involves three separate, but interconnected, areas of activity:

- Application of some perturbation to the heavy vehicle's wheels by some means or device which provides an excitation (forcing function) to induce dynamic behaviour;
- The instrumentation used for measurement of that dynamic behaviour; and
- The dynamic measures derived from that process.

Body bounce, axle hop and vehicle component forces as well as wheel forces may be derived from and by these either directly or by surrogate models.

The body of this report is divided into three broad areas. The instrumentation description is further broken down into two separate sections; the distinction between these being the instrumentation on or off the vehicle.

Accordingly, the divisions in this report are:

- equipment to apply perturbations (or forcing functions); discussed in Section 3 – "Evaluation of HV suspensions – forcing functions and associated equipment";
- the instrumentation used to measure the resultant dynamic parameters, discussed in:
  - Section 4 – "Evaluation of HV suspensions – on-vehicle instrumentation";
  - Section 5 – "Evaluation of HV suspensions – off-vehicle instrumentation"; and
- dynamic measures resulting from those processes; discussed in Section 6 - "Evaluation of HV suspensions – derived measures".
- Section 7 – "Findings" summarises the report content.

2 Background

2.1 Heavy vehicles & higher mass limits (HML)

Road authorities and transport regulators are under continuous pressure from the transport industry to allow “freight efficient” vehicles onto the road network. Outputs from the final report of the DIVINE project (OECD, 1998) were used in Australia to support the argument that air-sprung heavy vehicles (HVs) should carry greater mass under the micro-economic reform popular in the 1980s and 1990s in Australia. One of those reforms was the mass limits review (MLR) project as implemented under the 2nd heavy vehicle reform package (National Transport Commission, 2003). It was concluded that HVs operating at higher mass limits (HML) and equipped with “road friendly” suspensions (RFS) would be no more damaging than conventional heavy vehicles (HVs) operating at statutory mass with conventional steel springs (Pearson & Mass Limits Steering Committee, 1996). This resulted in the implementation of higher mass limits (HML) schemes in various guises in all Australian States.

HML allows HVs to carry greater mass in return for, amongst other requirements, being equipped with “road friendly” suspensions (RFS). This was the first indication that specific axle-mass increases had to be tied to vehicle design improvements. This was a clear signal from regulators that there would be little scope for any further blanket increases in GVM. This since the economic benefit from increases in GVM were not necessarily going to be balanced against the increasing costs of maintenance and capital for infrastructure capable of carrying heavier HVs. The road network asset had reached a point where any further gains in HV productivity would need to be traded off against more efficient boutique vehicles with improved design (Sweatman, 1994). Such incentives to encourage HV characteristics which did not consume the asset had been foreseen (Woodroofe, LeBlanc, & Papagiannakis, 1988). Details vary between Australian States in terms of HML access and conditions but, in terms of additional mass, HML generally allows increases above statutory mass of Δ2.5t on a HV tri-axle group and Δ0.5t on a HV tandem axle group.

The implementation of the various HML schemes in Australia has not stopped the road transport industry pressuring road authorities and transport regulators for more concessions on mass and vehicle combinations, however. The road transport industry's response to continued pressure from their clients for ever-increasing efficiency generally involves proposing heavy vehicles (HVs) towing more trailers with:

- a greater number of axles or axle groups;
- more gross vehicle mass (GVM);
- greater axle loadings; or
- greater axle group loadings.

Fewer prime-movers and drivers for a given freight task make these scenarios more attractive financially to transport operators and their clients. Accordingly, increasing numbers of HVs with more trailers, greater axle masses and axle group masses have been rolled out in response to such pressures. The first serious post-HML wave of these types of HVs is now operational although these vehicles have been on the network in various forms since the 1980s (Haldane, 2002) under the generic term "multi-combination vehicles" or MCVs. In an effort to minimise the asset damage from these non-standard HVs (including those operating at HML), regulators and road authorities continue to specify "road friendly" suspensions (RFS) as one of the conditions of access to the road network.

2.2 Higher mass limits & "road-friendly" suspensions

In 1984, German legislation allowed two-axle buses to carry an extra tonne on the rear axle. This was conditional on the axle-body interaction over a "step-test" having natural body-bounce frequency of less than 1.5Hz and a damping ratio of 0.25. The legislation relied on research which translated body-bounce parameters into road damage using a "road stress factor" (RSF) approach (Section 6.1.9) to determine damage equivalence. Accordingly, concessional allowances for vehicles with "road-friendly" suspensions were derived (OECD, 1992). These "road-friendly" suspensions incorporated air springs.

Air suspensions on HVs were then introduced more widely into Europe. This design-based blanket approval was made on the basis that an air suspension was installed. The background research to that move indicated that air-suspended drive axles at 11.5t produced the same road damage as non-air-suspended drive axles at 10.5t. The wider introduction of mass concessions did not, however, require similar performance or parametric tests to that specified for "road-friendliness" of HV suspensions in the German legislation (de Pont, 1992; Potter, Collop, Cole, & Cebon, 1994). Consequently in the UK in the early 1990s for instance, any air-sprung HV tri-axle group was allowed 24t; the same vehicles with non-air suspensions were restricted to 22.5t (de Pont, 1997).

In order that innovation not be stunted:

- by that design-based blanket approval (de Pont, 1997); and
- in developing "friendlier" non-air HV suspensions that were less damaging to road network assets (de Pont, 1992; Gyenes, Mitchell, & Phillips, 1994)

EC directive 85/3/EEC, as amended by Council directive 92/7/EEC (European Council, 1996), was introduced.

Similar to the German approach, a "load equivalence law" was used to formulate the EC directive. This to determine the amount of extra payload an air-sprung HV could be allowed if its damage effect was equivalent to that of a HV with conventional steel suspension at regulation mass (Collop & Cebon, 1997). The damage equivalence law used was based on that developed by Sweatman's (1983) "dynamic road stress factor" (Section 6.1.9) derived from Eisenmann (1975) to estimate pavement damage.

The directive allowed heavier axle loads for:

- air suspended drive axles in a blanket design-based approval; as well as
- other axles meeting the criteria and therefore approved as "air equivalent".

It specified a suspension testing regime for a suspension to be rated as "road-friendly" (or "air equivalent") provided:

- it had a natural body-bounce frequency of or below 2.0Hz;

- it had a damping ratio ≥ 0.2 (or 20%) of the critical damping value; and

- 50% or more of the viscous damping was provided by the shock absorbers.

These benchmarks were the criteria against which any suspension could have its "damage equivalence" measured. Any suspensions so rated were permitted the mass concessions afforded to air suspensions.

The HV suspension requirements under HML schemes in Australia broadly followed the European assumption that air suspensions on HVs should be allowed a payload advantage over conventional steel-sprung axles based on "damage equivalence". Australia incorporated the 92/7/EEC parameters and tests into its VSB 11 certification regime for road friendly suspensions as well as adding additional requirements regarding static load-sharing (Appendix 3). RFS in Australia generally incorporates air springs, although there are some steel-sprung RFS emerging onto the market (Australia Department of Transport and Regional Services, 2004b).

It is for noting that the body-bounce frequency was not necessarily the only available choice to determine well-behaved suspensions when the EU and, later, Australian specifications for "road friendly" suspensions were formulated. HVs have their vehicle body bounce frequencies in the range 1.2 – 4Hz whilst the axle bounce (axle hop) frequencies are in the 8 – 18Hz range (Gyenes & Simmons, 1994; Middleton & Rhodes, 1991; Mitchell, 1987; OECD, 1992). Further, of the two dominant responses of HVs to suspension perturbations, axle-hop has the greater magnitude forces (Sweatman & Addis, 1998).

The RFS standard in Australia is regulated by the Department of Transport and Regional Services (DoTaRS) (2004a). DoTaRS certifies "road friendly" suspensions under testing defined in the Vehicle Standards Bulletin No. 11 (Australia Department of Transport and Regional Services, 2004a). As with the EC directive based on "air-equivalence" of RFS suspensions (de Pont, 1992), two of the parameters which are used to define HV "road-friendliness" in Vehicle Standards Bulletin No. 11 (VSB 11) are the damping ratio and natural damped frequency of an axle/body assembly, singly or of axles within a group.

HV axle models are certified as "road-friendly" via a "type test" of a representative sample of a model of an axle either singly or as a group. Certified axle/s fitted to any particular HV when it is first registered for HML or other duties requiring RFS deems that HV to be "road friendly". Hence, RFS-equipped HVs in Australia are certified to VSB 11 as "road friendly" at the time of manufacture and with type-tested axles.

2.3 Air suspensions *vs.* conventional steel – design and performance differences

The majority of conventional steel-spring suspensions on modern HVs are provided with shock absorbers. Should these shock absorbers wear, residual damping is provided via hysteretic Coulomb friction between the leaves of the springs (Prem, George, & McLean, 1998). Air suspensions with worn shock absorbers have no equivalent alternate form of damping. This because air springs have no similar damping characteristic to that of conventional steel suspensions. Further, conventional HV leaf springs exhibit so much Coulomb friction for most undulations in the road surface that they barely move; surface undulations being accommodated by tyre flexure (Dickerson & Mace, 1981; OECD, 1992). Studies have shown low friction liners between leaves in conventional HV springs can relieve this behaviour (Simmons & Wood, 1990).

After measuring dynamic HV wheel loads with both wheel and pavement instrumentation, Hahn, p8 (Hahn, 1987a) noted that "A modern parabolic-spring guided lorry axle with hydraulic shock absorbers does not generate significantly greater dynamic wheel forces than does an air-suspended trailer axle."

The research reports documenting testing for wheel forces on a length of the A34 in the UK (Potter, Cebon, Cole, & Collop, 1995) noted that air suspensions fitted to prime-mover drive axles increased wheel forces for all trailer suspension types when compared with steel-suspended prime-movers. Further, that study noted that air suspensions fitted to prime-mover drive axles often caused higher whole-of-vehicle road damage than that caused by steel-suspended prime-movers. It was not clear whether these air suspensions were equipped with dampers and/or met EC directive 92/7/EEC.

Prior to 1999, all air-suspended HVs were "deemed" "road-friendly" by Australian road authorities, an example of blanket "type-approval" for HV suspension design. de Pont tested a particular type of air suspension that proved not to meet the EC damping ratio and frequency requirements. That report noted, p52 (de Pont, 1997):

"… [the tested] air suspension did not generate lower levels of dynamic loading than the [tested] steel suspension and thus cannot be rated as more 'road friendly'. Although the suspension was softer, it was inadequately damped. This highlights the critical importance of damping on the performance of softer suspensions and shows the weakness of a design-based approach to 'road-friendliness' rating. The poor damping performance of this suspension was completely attributable to poor design. All the components in this suspension were in good condition and performing to specification…this particular type of suspension is widely used in New Zealand". Accordingly, that work supported an argument against the design-based blanket approval approach for particular types of suspensions (de Pont, 1997).

VSB 11 was brought in as a requirement after 1999 (Australia Department of Transport and Regional Services, 2004a). A significant proportion (approx. 30%) of air-suspended semi-trailers on the Australian road network were in operation prior to any of them being required to meet VSB 11 (Blanksby *et al.*, 2006). Semi-trailers have a working life of up to 30 years. The legacy of no in-service testing and the decision to "deem" air suspensions as RFS without design type-testing is an on-going presence of HVs with "grandfathered" approval for air suspensions operating on the road network. This issue was recognised by Starrs *et al*., (2000) as a significant problem regarding any in-service RFS HV testing proposal since "grandfathered" suspensions:

- may not pass in-service tests; and
- have been on the road since before "road-friendly" requirements were introduced.

The Starrs *et al*., (2000) report estimated that these 'grandfathered' suspensions would be phased out economically in 5 years but this has not eventuated (Blanksby *et al*., 2006).

The final report of the DIVINE project, p77 (OECD, 1998) noted, authors' italics for emphasis:

"When air-suspended vehicles travelled at critical speeds over axle-hop inducing features, large dynamic responses and multiple fatigue cycles were observed. These responses were *up to 4.5 times the dynamic load allowance* specified in bridge design."

Hence the original study used to justify HVs with RFS at HML loads acknowledged that wheel forces from HVs with air suspensions could impose high dynamic loads on infrastructure. However, the issue of higher dynamic loadings arising from air-sprung suspensions was not adequately dealt with in the final report of the DIVINE project (OECD, 1998) according to Pesterev *et al.*, (2004). That study explored the issue of dynamic loadings of air-sprung HVs with respect to conventional steel suspensions. It noted that:

- the final report of the DIVINE project defined air suspension to be more "road-friendly" than steel suspension; and
- its recommendation [was] for the use of these instead of the conventional steel suspensions (Pesterev *et al.*, 2004),

and went on to examine the statement, p77 (OECD, 1998):

"For short-span bridges with poor profiles, large dynamic responses occur for both air- and steel-suspended vehicles. The highest measured responses were for short-span bridges with poor damping that are traversed by air-suspended vehicles with specifically-excited axle-hop vibrations."

Pesterev *et al.*, (2004) then modelled air and conventional steel suspensions for body-bounce and axle hop over short-span bridges. This showed a reduction in forces associated with the body-bounce for air suspensions but also found a resultant increase of approximately 15% in the magnitude of higher-frequency axle-hop forces.

Pesterev *et al.*, p18 (2004) concluded (authors' italics):

"...an air-suspended vehicle is potentially dangerous for short-span bridges with fundamental frequencies in the range of vehicle axle-hop frequencies. Moreover, although air-suspended vehicles are considered ''road-friendly'', *they can produce a greater pavement damage* compared to steel-suspended vehicles in the case of uneven road surface with short-wavelength irregularities, which excite the axle-hop vibration. We believe that the above explanation of the phenomenon discussed is more realistic (and simpler) than that given in the DIVINE report…"

2.4 The contribution damping makes to the "road-friendliness" of HV suspensions

HV air suspensions rely on suspension dampers (shock absorbers) almost entirely for damping. The action of conventional shock absorber damping is termed "viscous damping" and is dependant on velocity and force (Prem *et al*., 1998; Sweatman *et al*., 2000). Shock absorbers control the speed of movement between the axle and the body of a vehicle and convert the energy generated thereby into heat that is dissipated by the outer case of the damper.

Dynamic forces generated by typical travel over commonplace unevenness in roads produce very high dynamic loads when RFS are lightly damped compared with those generated by conventional HV steel suspensions (Costanzi & Cebon, 2005; OECD, 1998).

The final report of the DIVINE project, p212 (OECD, 1998) stated:

"Although it has been suggested, for example, that air suspensions with worn shock absorbers may have no advantage over steel spring suspensions and may actually create higher dynamic loading under some circumstances, there is little current evidence to support this claim".

This view was re-stated in later work (Sweatman *et al.*, 2000) where it was claimed that, in a worst-case scenario and should shock absorbers be rendered totally ineffective, the dynamic loadings due to air suspensions would increase only to similar levels imposed by non-road-friendly mechanical suspensions. That report concluded that degraded air suspensions would cause no higher dynamic loading than conventional steel suspensions.

There were other findings available to the DIVINE project team that showed the results of ineffective shock absorbers on HV wheel loadings, however. Woodrooffe (1996) reported on dynamic loading tests where the wheels of a loaded HV were subject to a 1mm sinusoidal sweep frequency input. Comparing the test case for dampers in good condition with the case of ineffective shock absorbers, a summary of peak dynamic loading during these tests at the body's and the axle's resonant frequencies may be summarised thus:

> Let the dynamic wheel-force magnitude for body-bounce *and* axle-hop of a conventional steel tandem (2-axle) HV suspension both be normalised to 1.0 at their eigenfrequencies.
>
> Tandem air suspension with shock absorbers:
>
> - dynamic wheel-force magnitude due to body-bounce: 0.14
> - dynamic wheel-force magnitude due to axle-hop: 0.93
>
> Tandem air suspension without shock absorbers:
>
> - dynamic wheel-force magnitude due to body-bounce: 0.29
> - dynamic wheel-force magnitude due to axle-hop: 3.5
>
> (Woodroofe, 1996).

For the two test cases (dampers in good condition *vs.* ineffective dampers), other studies have found peak dynamic loading for similar tests summarised thus:

> Again, let the normalised dynamic wheel-force magnitude for body-bounce *and* axle-hop of a conventional steel tandem (2-axle) suspension at their eigenfrequencies = 1.0.
>
> Tandem air suspension with shock absorbers:
>
> - dynamic wheel-force magnitude due to body-bounce: 0.11

- dynamic wheel-force magnitude due to axle-hop: 7.5

Tandem air suspension without shock absorbers:

- dynamic wheel-force magnitude due to body-bounce: 0.28
- dynamic wheel-force magnitude due to axle-hop: 24.8

(Sweatman & Addis, 1998).

That study also quoted testing where the dynamic loading was reported to have increased by 110% when the shock absorbers were removed from one side of an air-suspended tri-axle group (Sweatman & Addis, 1998).

The final report of the DIVINE project (OECD, 1998) showed that dynamic wheel loads can increase by Δ50kN when an air suspension has ineffective, damaged or out-of-specification shock absorbers.

When compared with conventional steel suspensions of the day, the lower spring constants and higher damping ratios of air suspensions contributed to a view that air suspensions were less damaging to the network asset. Around the time that air suspensions were being implemented in Europe, Australian regulators and road authorities were considering mass concessions in return for air-suspended HVs (de Pont, 1992).

One of the major outcomes of the MLR project (National Road Transport Commission, 1993b) was that air-sprung HVs with RFS (i.e. within the parametric limits proposed and with functional dampers) should be allowed to operate at higher masses (HML) if they did not damage the roads any more than conventional steel suspensions at statutory masses. This outcome was heavily dependant on the correct operation of shock absorbers.

The differing views as to the "friendliness" of RFS beg the question, then, regarding the damage done by HVs with air suspensions with out-of-specification or deficient shock absorbers. This was addressed in the work of Costanzi and Cebon eight years after DIVINE and will be covered in Section 2.7 (Costanzi & Cebon, 2005).

2.5 Testing HV suspension "road-friendliness" under VSB 11

As seen later (Section 2.9), 2nd order systems (such as HV suspensions) may be characterised for their damping ratio and damped natural frequency (also called, not entirely correctly, "fundamental frequency" or just "frequency") by subjecting a system to a perturbation (forcing function) and measuring the response as an output. Two measurements used to show that heavy vehicle suspensions are "road friendly" are the body-bounce damping ratio and the body-bounce damped free vibration (natural) frequency. Normally these are found by expensive laboratory testing where a representative axle assembly is installed on a test (or surrogate) vehicle. This is mounted on a frame or hoisted. The test "vehicle" is then subjected to calibrated jolts or vibrations with its dynamic axle-to-body bounce movement measured and analysed.

The overall purpose of developing and applying evaluation &/or testing of RFS is to ensure that HVs thus equipped meet VSB 11 parameters when entering normal service.

Vehicle Standards Bulletin No. 11 (VSB 11) (Australia Department of Transport and Regional Services, 2004a) specifies requirements when a suspension is subject to specified tests. These parameters are influenced by the health of the vehicle's suspension dampers. VSB 11 certification requirements may be applied to an axle singly or to axle groups. The tests are, in brief (for detail, see Appendix 3):

- a pull down test where the chassis is pulled down by some means and then released;
- a drop test where the chassis is pulled up and released suddenly; and
- a step-down test where the vehicle is driven over an 80mm step-down profile.

The viscous damping (i.e. damping provided by the shock absorbers) needs to be more than 50% of all damping in the system. This requirement is usually determined by testing with and without shock absorbers. The damping ratio for the tests with shock absorbers is then compared to the damping ratio results for the tests without shock absorbers; the former needs to be greater than 50% of the latter for the test to be passed satisfactorily.

The damping ratio and body-bounce frequency parameters are measured at full load in single (10t), tandem (8.5t) or tri-axle (7.5t) configurations (Australia Department of Transport and Regional Services, 2004a).

In practice, axles are not necessarily coupled in groups. Where the same single axle is used it is tested at least 3 times with axle loads corresponding to those it would encounter in those different single, tandem or tri configurations (Bisitecniks, 2007).

Load-sharing is a requirement of VSB 11. Acceptable load-sharing is defined as all static axle loads when fully loaded being within 5% of one another as well as, somewhat confusingly, all wheel loads within 5% of each other.

An axle or axle group meeting all of the foregoing requirements may then be certified as “road friendly”. A certificate is then issued for each different mass and axle configuration e.g. single, tandem or tri, for that model of axle.

2.6 “Road-friendly” suspensions – the search for in-service testing

In Australia at present, there are no specific requirements for HVs operating at HML loads to have their RFS tested to VSB 11 once the HV is in service. This despite the original MLR project containing authoritative statements regarding the nexus between road network asset damage, damper health and friendliness of air suspensions (National Road Transport Commission, 1993b; OECD, 1998).

Gyenes *et al*., (1992) examined the issue of road damage, maintenance costs due to conventional steel suspensions *vs.* air suspensions and testing for “road-friendliness”. That study proposed a number of methods for managing HV “road-friendliness” both in-service and when new. Amongst these were:

- blanket design approvals according to springing media;
- parametric testing of body-bounce after either step-down, drop or frequency scan on a road simulator (Section 3.2.1);
- simulation testing; or

- testing over artificial road profiles.

Cebon noted similar issues and methods. Type-testing of new suspensions against a defined standard, backed up by annual inspections incorporating in-service tests on all air-suspended vehicles, was recommended (Cebon, 1999).

In-service testing was supported in theory by two further studies after the implementation of HML in Australia, those of Sweatman *et al.*, (2000) and Starrs *et al.*, (2000). Sweatman *et al.*, (2000) investigated in-service suspension testing techniques. That report to the NRTC (as it was then) covered:

- shock absorber testing machines and methods;
- shock absorber characteristics;
- shock absorber wear;
- suspension performance; and
- any concomitant degradation due to that wear.

A lower bound of 15% (0.15) for damping ratio determined from in-service testing was recommended. Sweatman *et al.*, (2000) also suggested an alternative method for maintaining RFS health: regular replacement of suspension dampers as part of the normal maintenance discipline in the transport operator's workshop. That report went on to note that governments (i.e. regulators and road authorities) were requiring the development of low-cost in-service testing methods that were able to determine body-bounce frequency, damping ratio and load-sharing.

The frequency and damping ratio parameters defined in VSB 11 are determined for axle-to-body movement. Accordingly, Sweatman *et al.*, (2000) discussed the requirement for any in-service test to have instrumentation which measured these parameters and eliminated (or reduced to an acceptable level) any contribution that tyre dynamics made to body bounce. Starrs (2000) took the outputs of the Sweatman *et al.*, (2000) report and examined the economics of shock absorber replacement and the use of tests as defined by the final report of the DIVINE project (OECD, 1998). The Starrs work also explored HV suspension evaluation and maintenance in a general manner. Neither report proposed a definitive way forward (Starrs Pty Ltd *et al*., 2000; Sweatman *et al*., 2000).

At the time of the Starrs and Sweatman studies the estimated savings on pavement rehabilitation in the case of all RFS operating correctly was $14M across the Australian HV network. That quantum was not considered to be cost-effective within the context of total pavement rehabilitation costs and operational costs to test RFS HVs (Starrs Pty Ltd *et al*., 2000). It is noted that one of the authors of the Sweatman *et al.,* (2000) report had already recommended, at the international level, type-testing of RFS using parametric or other means combined with annual in-service testing. That work detailed extensively the options for the associated test equipment needed for these measures in a regulatory framework (Cebon, 1999). Others (Potter *et al*., 1997; Woodroofe, 1995), agreed. Even earlier, the need to test new-generation HVs for characteristics which contributed to their "road-friendliness" was recognised as "probable" (Woodroofe *et al*., 1988).

Despite these efforts and the case in favour of in-service testing there is still no requirement for in-service testing of HV RFS parameters and no concomitant in-service test recognised in Australia. Such is the concern surrounding the issue of network asset damage from air-suspended HVs with deficient suspension dampers that two Australian States (Australia Department of Transport and Regional Services, 2005a, 2005b) have, in conjunction with the Australian Government, included conditional clauses in their current Bilateral Infrastructure Funding Agreements (BIFAs). Extracts of these are reproduced in Appendix 2. National activity on in-service HV suspension testing in Australia lapsed from the time of the Starrs and Sweatman studies until NSW's and Queensland's BIFA negotiations in 2004 – 2005.

2.7 "Road-friendly" suspensions – the case for in-service testing

Within the framework described above and its indeterminacy with respect to RFS health, regulators and road authorities have not been able to be certain that air-sprung heavy vehicles (HVs) with RFS are having their "road friendliness" maintained as the suspension dampers wear from normal service. Hence a scenario of non-standard MCVs with more than statutory mass (at or higher than HML loadings) on axles or axle groups with worn suspension dampers is possible. Of course, if transport operators were maintaining their vehicles to specification and regulation then there would be no need for concern on the part of road authorities and transport regulators. However, recent work in NSW has indicated (Blanksby *et al.*, 2006) that 54% of HVs in a statistically valid survey did not meet at least one of the requirements of VSB 11. The possible scenario of non-standard MCVs with more than statutory mass (at or higher than HML loadings) on axles or axle groups with worn or out-of-specification suspension dampers has now become a better than even-money probability. This is of concern to Australian road authorities and transport regulators.

Costanzi & Cebon (2005) modelled pavement rehabilitation fleet of HVs with varying levels of ineffective dampers, one scenario was for a HV fleet where 50% of that fleet had “poorly-maintained” dampers. That report concluded that, at Higher Mass Limits loadings, pavement and surfacing damage would be 20 - 30% greater than for a comparable freight task with a fleet equipped with dampers in good condition. The Costanzi & Cebon study was for HVs on the Newell Highway. The Newell has considerably thicker pavements than those found in Queensland (Queensland Department of Main Roads, 2007b). If the figure of 50% poorly maintained suspensions, modelled by Costanzi *et al*., is equated to the actual status found at Marulan then a HV fleet with 100% functional shock absorbers would save Queensland Main Roads’ maintenance budget in excess of $59M/annum in 2007 dollars (Queensland Department of Main Roads, 2007a); a saving going forward every year. This without considering the comparative fragility of the Queensland network to higher loads (with respect to the Newell’s >200mm thick pavements). Savings such as these are essentially “free money” since HV suspensions should be maintained as a matter of course. This also indicates that the previously estimated benefit to pavement rehabilitation costs of $14M (Starrs Pty Ltd *et al*., 2000) was low, even allowing for cost escalation and inflation with the effluxion of time. Further, the consideration of well-maintained dampers does not include the road safety or workplace health and safety aspects of a HV fleet with greater than 50% sub-optimal shock absorbers.

2.8 Dynamic and static load-sharing in air-sprung HVs

Load-sharing within a suspension group can be defined as the amount of equalisation of the axle group load across all wheels/axles in a multi-axle group. A variation on that definition is that a HV with a "load equalising system" means that, p26 (Stevenson & Fry, 1976):

- an axle group utilises a suspension with the same spring types on each axle; and
- the design delivers "substantially equal sharing by all the ground contact surfaces of the total load carried by that axle group".

Soon after this study, early efforts to define "load-sharing" in Australia were made (Australia Department of Transport, 1979).

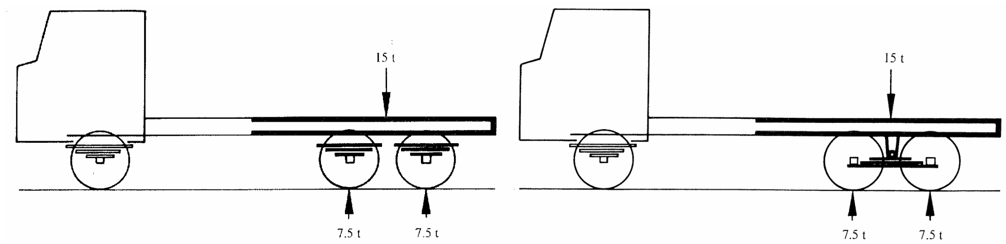


Figure 1. Early attempts to define load-sharing (Australia Department of Transport, 1979). The suspension on the left was defined as non-load-sharing because of effect shown in Figure 2.

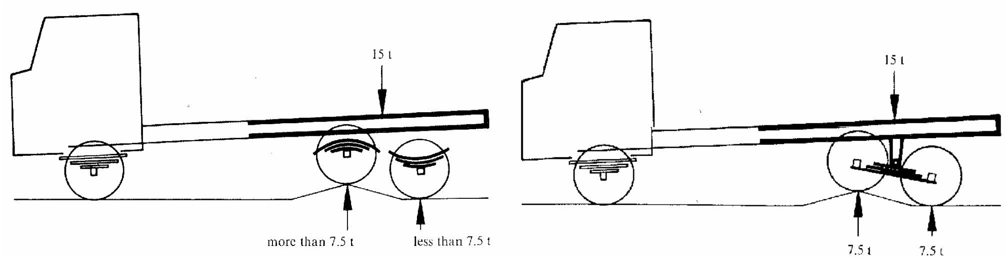


Figure 2. Showing the effect of wheel-forces with non-load-sharing suspension, left *vs*. load-sharing suspension, right (Australia Department of Transport, 1979).

The suspension on the right in Figure 1 & Figure 2 is an example of a centrally pivoted suspension although the one shown is not the only expression of this design. It is apparent from Figure 1 & Figure 2 that load-sharing was seen at the time to be a static or quasi-static phenomenon. This was recognised by Sweatman (1983) as only part of the issue. That report as well as others (Cole & Cebon, 1991) contended centrally-pivoted suspensions with inadequate damping by design would be less "road-friendly" (Sweatman, 1983). This because underdamped transmission of front-axle perturbations to the rear axle via the rocker-arm mechanism would lead to high wheel forces.

The final report of the DIVINE project nominated suspension load equalisation as being important to "road-friendliness" and recommended the following measure, p107 (OECD, 1998):

"Load equalisation may be evaluated on the basis of average load variation per unit of relative vertical axle displacement (for example, 100 mm of travel)... To qualify as a road-friendly tandem suspension, it is recommended that differential axle load variation must be no greater than 0.3 kN/mm based on a 9 tonne axle load…"

Again, this was a quasi-static approach and did not address dynamic equalisation in that a time constant or period for equalisation was not specified, only a differential value of deflection.

The overarching load-sharing requirement (independent of suspension type) for HVs in Australia is set down in the Australian Vehicle Standards Rules 1999 in which Rule 65 is as follows (Australia Parliament, 1999):

"**65 Relation between axles in axle group**

(1) The axles in an axle group, except a twin steer axle group, fitted to a vehicle with a GVM over 4.5 tonnes must relate to each other through a load-sharing suspension system.

(2) In this rule:

load-sharing suspension system means an axle group suspension system that:

(a) is built to divide the load between the tyres on the group so that no tyre carries a mass over 10% more than the mass that it would carry if the load were divided equally; and

(b) has effective damping characteristics on all axles of the group…"

Early experiments, p6 (Sweatman, 1976) concluded that, when it came to load-sharing, "…there appears to be little correlation between static and dynamic suspension performance".

The DIVINE project final technical report, p77 (OECD, 1998) found that air-suspended HVs do not load share in the dynamic sense (authors' italics for emphasis):

> "When air-suspended vehicles travelled at critical speeds over axle-hop inducing features, large dynamic responses and multiple fatigue cycles were observed. These responses were up to 4.5 times the dynamic load allowance specified in bridge design. Where axle hop was not induced, the dynamic response was much smaller. A probable explanation for this is the fact that the *very limited dynamic load-sharing in air suspensions* allows the axles in a group to vibrate in phase at axle-hop frequencies. "Crosstalk" between conventional steel leaf suspensions limits this possibility. This difference in behaviour was crucial in the strength of the dynamic coupling between air suspensions and short-span bridges."

Earlier OECD work (OECD, 1992) noted that an inequality in dynamic load-sharing between axles of 20% increased road damage by a factor between 1.2 to 3.0. When implementing RFS requirements in Australia, a 5% imbalance between axles or wheels in a static test was seen to overcome this concern (National Road Transport Commission, 1993a). This requirement became part of VSB 11 as part of its static test outcomes. With respect to load-sharing or load equalisation, the VSB 11 specification, p8 (Australia Department of Transport and Regional Services, 2004a) nominates that RFS suspensions must meet the following requirement: "Static load share between axles in the axle group must be within 5%".

There is an anomaly on p9 however, in that a load-sharing suspension is defined thus:

"**load-sharing suspension system** means an axle group suspension system that:

(a) is built to divide the load between the tyres on the group so that no tyre carries a mass more than 5% greater than the mass it would carry if the load were divided equally…"

Hence, one requirement for load-sharing is applied to axles and another requirement defines load-sharing between wheels. Further, neither VSB 11 nor any other official document defines a formal methodology to determine a static load-sharing value (Prem, Mai, & Brusza, 2006). That detail has been left to a method suggested in a monograph (official status unknown) issued by Mr. Wong of the Australian Department of Transport and Regional Services.

Dynamic load-sharing can be defined as a measure of the ability of a HV multi-axle group to equalise load across its wheels under typical travel conditions of a HV; i.e. in the dynamic sense at typical travel speeds and operating conditions of that vehicle. Sweatman (1983) attempted to quantify the load-sharing ability of a multi-axle group in a number of ways, including the concept of "load-sharing coefficient" (LSC). Potter *et al.*, (1996) clarified various methods for quantitative derivation of measures to describe the ability of an axle group to distribute the total axle group load during travel. Despite this, there is no agreed testing procedure to define or measure dynamic load-sharing at the local or national level in Australia.

2.9 Theory of determining parameters of 2nd order systems *viz:* HV suspensions

Characterising heavy vehicle (HV) suspensions is central to the EU (and Australian) test for "road friendliness" (Australia Department of Transport and Regional Services, 2004a; European Council, 1996). Two measurements used to show that heavy vehicle suspensions are road friendly are the damping ratio, zeta (ζ) and the damped free vibration frequency (f).

The damped free vibration body-bounce frequency is the frequency at which a truck's body has a tendency to bounce on its suspension with the largest excursions whilst being damped by the suspension dampers (shock absorbers).

The damping ratio is a measure of how fast a system reduces its oscillations (and returns to quiescent or steady state) after a disturbance. It is a measure of the reduction in excursions of subsequent amplitudes of the output signal from a 2nd order system. In HV suspensions, it is related to a measure of how quickly the shock absorbers and other components reduce body bounce and wheel hop after the truck hits a bump. The damping ratio, zeta (ζ), is a dimensionless number and is usually presented as a value under 1 (e.g. 0.3) or a percentage (e.g. 30%) denoting the damping present in the system as a fraction of the critical damping value (Doebelin, 1980).

Chesmond (1982) showed that system parameters may be characterised in a number of ways. Amongst these were:

- application of a random input signal to a system. Fourier (or other frequency domain) analysis of the output signal resulting from that random input may be used to determine the characteristics of the system transfer function. The damped free vibration frequency (f) of the system characterises that transfer function and will show up as the largest magnitude peak in the frequency spectrum of the output signal; or

- application of an impulse input signal to a system: a perfect impulse signal contains all frequencies in equal proportion. Again, Fourier analysis of the output signal may be used to determine the characteristics of the system transfer function. Similar to random input signal excitation, the dominant (in this case, damped free vibration) frequency will manifest as the largest peak in the frequency spectrum of the output signal for a given impulse input.

- Subjecting any system to an impulse signal and measuring the reducing excursions of the output signal enables the damping ratio of a system to be determined.

The input signal used to excite the system is known as the forcing function.

Doebelin, p79 (1980) said, on the subject of length of time over which the impulse function is applied and its shape: 'We see that if [the input function's] duration is "short enough", the system responds in essentially the same way as it would to a perfect impulse of like area and that *the shape of* [the input function] *makes no difference whatsoever*.'

The damping ratio (ζ) may be determined by comparing the values of any two consecutive response peaks in the same phase (i.e. comparing the magnitudes of the 1st and 3rd excursions or the 2nd and 4th excursions) of the output signal of an underdamped 2nd order system after an impulse function input has been applied

(Meriam & Kraige, 1993). Prem, *et al.*, (2001) used the formula (Meriam & Kraige, 1993):

$$\zeta = \frac{\delta}{\sqrt{\delta^2 + (2\pi)^2}}$$

Equation 1

where δ is the standard logarithmic decrement (Meriam & Kraige, 1993) given by the following formula:

$$\delta = \ln\left(\frac{A_1}{A_2}\right)$$

Equation 2

Where:

A_1 = the amplitude of the *first* peak of the response; and

A_2 = the amplitude of the *third* peak of the response

to determine damping ratio.

Alternatively, A_1 and A_2 may be described as the first two peaks of the response that are in the same direction (i.e. on the same side of the x-axis of the time-series signal of the response).

Where it is desired to determine damping ratio from a signal with more than 2 peak values of the signal on the same side of the x-axis in a time series, Thomson & Dahleh (1998) provide:

$$\delta = \frac{1}{n}\ln\left(\frac{x_0}{x_n}\right)$$

Equation 3

to substitute into Equation 1, where:

x_n = the amplitude after *n* successive cycles have elapsed; and

x_0 = the amplitude when n = 0; or

for the case where continuous successive peaks are present (Technical Committee ISO/TC 22, 2000):

$$\delta = \ln\left[\frac{1}{n-1}\left(\frac{x_1}{x_2}+\frac{x_2}{x_3}+......+\frac{x_n}{x_{n+1}}\right)\right]$$

Equation 4

Note: Equation 1 may be derived by solving for ζ in the following equation (Meriam & Kraige, 1993) as shown in Appendix 4:

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}$$

Equation 5

2.10 Sampling theory – Shannon's theorem and Nyquist criterion

In order to re-create a signal with frequency components from 0Hz to f_i Hz,

Where:

f_i Hz is the maximum frequency of interest;

the Shannon-Nyquist sampling theorem states that the sampling rate must be a minimum of 2 f_i Hz (Considine, 1985).

Noting that frequency ω in radians is related to conventional frequency f in Hz by:

$$\omega = 2\pi f$$

Equation 6

the following proof may be considered.

Let T be the system sampling time in seconds (s) of a continuous time-series signal where the highest frequency of interest in the sampled signal is ω_i rad/s. The continuous time-series signal may be represented by:

$$e(t) = \sin(\omega_i t + \theta)$$

Equation 7

Let the sampling frequency in radians be ω_s where $0 < \omega_i < \omega_s/2$.

The sampling frequency ω_S relates to conventional sampling frequency f_S in Hz by:

$2\pi f_S = \omega_S = 2\pi/T$.

=> $\omega_S = 2\pi/T$

=> $T = 2\pi/\omega_S$

Equation 8

Let $t = kT$ where k is a constant.

Substituting kT for t in Equation 7, the equation for the sampled signal may be represented:

=> $$e(kT) = \sin(k\omega_i T + \theta)$$

Equation 9

Now consider a different continuous time-series signal with higher frequencies where the higher frequencies contain a component $n\omega_S$ in addition to ω_S and represented by:

$$f(t) = \sin\left[(\omega_i + n\omega_s)t + \theta\right]$$

Equation 10

Where $n = 1, 2, 3\ldots$

from Equation 8:

$\omega_S = 2\pi/T$

Substituting $2\pi/T$ for ω_s from Equation 8, Equation 10 becomes:

$$f(t) = \sin\left[(\omega_i + \frac{2\pi n}{T})t + \theta\right]$$

Equation 11

For which the sampling time T is as above.

Substituting kT for t as before, the equation for the reconstructed signal is then:

$$f(kT) = \sin\left[(\omega_i + \frac{2\pi n}{T})kT + \theta\right]$$

=> $$f(kT) = \sin(kT\omega_i + \theta + 2\pi kn)$$

=> $$f(kT) = \sin(kT\omega_i + \theta)$$

Equation 12

As the signal frequencies in Equation 11 increase by a sum proportional to $2\pi/T$, the reconstructed signal in Equation 12 becomes indistinguishable from Equation 9, even though the frequency of the signal has been increased.

Hence $\omega_i < \omega_s/2$ is the criterion for the process of regular time-based sampling. As

$$\omega_i \underrightarrow{\lim} n\omega_s/2$$

the reconstruction process for sampled signals is unable to make the distinction between any two sampled signals for all cases of integer values of n. Beyond this limit, where $\omega_i \geq \omega_s/2$, the reconstructed frequencies are "folded" back into the frequency spectrum interval $0 < \omega_i < \omega_s/2$ and it becomes impossible to reconstruct ω_i frequencies greater than $\omega_s/2$ from the sampled signal. The critical sampling frequency π/T is often referred to as the "Nyquist frequency". The selection of small enough values of T to achieve valid reconstruction of signals is termed Shannon's theorem, which may be stated:

$$\omega_s > 2\,\omega_i \text{ or } \pi/T > \omega_i$$

(Houpis & Lamont, 1985).

3 Evaluation of HV suspensions - forcing functions and associated equipment

3.1 Dedicated test inputs and associated equipment

3.1.1 General

As the standard for RFS in Australia, VSB 11 allows a variety of test methods to determine the suspension parameters for damped free vibration body-bounce frequency and damping ratio. Some of these test methods are defined and others are unspecified but permitted, subject to DoTaRS satisfaction. One option under VSB 11 (Australia Department of Transport and Regional Services, 2004a) and the EC (1996) tests for "road friendliness" is the use of a step-down test, the others are "pull-down and release" or "pull up and drop" (Appendix 3). These methods lend themselves to the design and manufacture of test machinery that impart just the one type of forcing function. These methods and associated machinery are covered in this section.

3.1.2 Drop test as forcing function

Gyenes *et al.,* (1992) summarised the possibilities for testing suspensions using "parametric testing" where the parameters of damping ratio and body-bounce frequency are determined since they correlate with the dynamic response of a suspension. One of the methods suggested was the "lift-and drop" method also known as "pull-up-and-drop" or "drop test" method. This method is included in VSB 11 and EC test procedures. It is a forcing function that will cause a perturbation in the HV suspension such that the natural body-bounce frequency and damping ratio may be derived from the resultant transient oscillations as measured by appropriate transducers. This test is used in Australia by Bisitecniks for certifying original equipment axles for RFS by raising the test vehicle with an overhead crane or a collapsing frame under the chassis (Bisitecniks, 2007).

By dropping a vehicle a short distance using collapsing supports, Milliken *et al.*, (2001) developed a reliable test to determine damping ratio and natural body-bounce frequency. This test method was designed to replicate the EC test for RFS by exciting the suspension and allowed measurement of its transient response. The mechanism for dropping the truck was a low frame placed under the wheels of an axle similar to that shown in Figure 3 and Figure 4 (Transfund New Zealand, 2001).

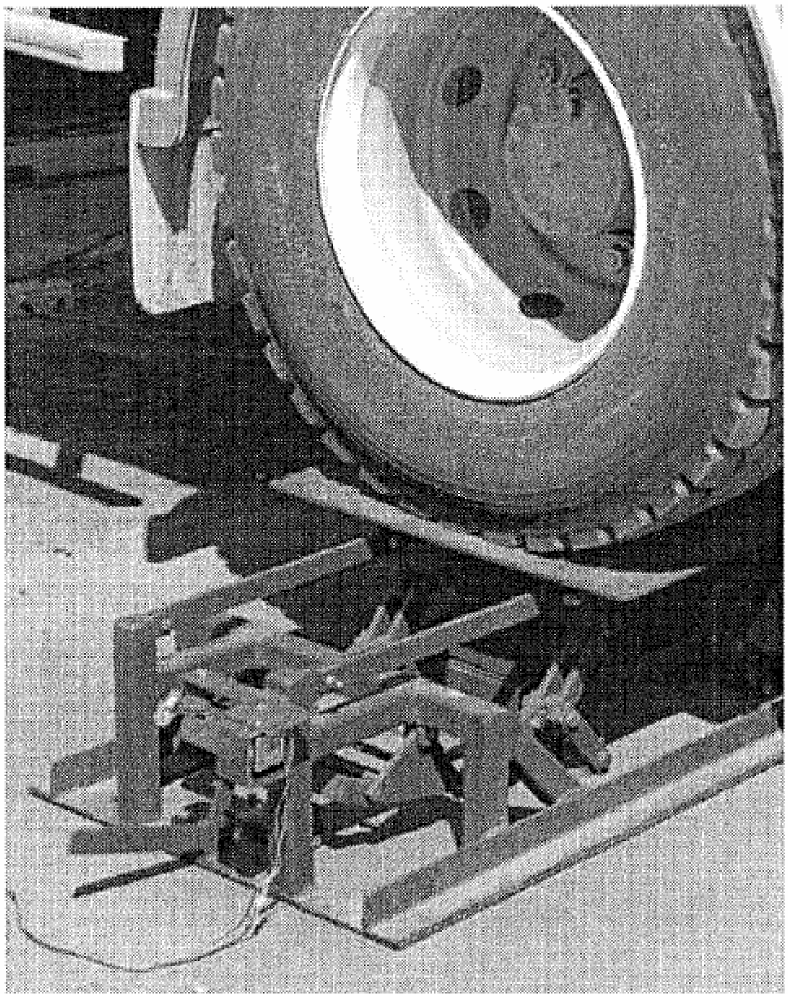


Figure 3. Detail of the collapsing frame used in the Transfund New Zealand suspension testing apparatus (Transfund New Zealand, 2001).

It was configured so that both sides of the frame would collapse simultaneously. Three different values of drop heights were used: 48mm, 80mm and 112mm. No significant differences in the measured suspension parameters were apparent for different drop heights. Milliken *et al.,* (2001) noted that the 112mm height was unnecessarily vigorous. This method was developed into a commercial application (Transfund New Zealand, 2001). Accuracy was reported to be ±7% for frequency and ±18% for damping ratio. As noted by others (Woodroofe, 1995), damping ratio is usually the most difficult suspension parameter to measure accurately in that shock absorbers act differently depending on direction of travel (Uffelmann & Walter, 1994).

Figure 4. General arrangement of the Transfund New Zealand suspension testing apparatus showing application to single-drive axle rigid truck (Transfund New Zealand, 2001).

The ARTSA air suspension code (Australian Road Transport Suppliers Association, 2001) references a drop test rig. This was developed by Dr. Sweatman and Roaduser. ARTSA quotes the EC test in their 2001 booklet. Figure 5 shows the patented Sweatman apparatus. The test vehicle is lifted using pneumatic actuators that are then exhausted via a poppet valve. This action drops the vehicle from a height of 80mm (Australian Road Transport Suppliers Association, 2001). The guide notes that the subsequent oscillations may then be analysed. This rig was used by Blanksby *et al.*, (Blanksby *et al.*, 2006) for the Marulan survey. No details of the accuracy or precision for this method are available. In 2001, Dr. Sweatman was the Chairman of ARTSA.

Figure 5. ARRB (ex-Roaduser) drop tester in action

3.1.3 Step-down (step test)

Another approved testing method under VSB 11 (Australia Department of Transport and Regional Services, 2004a) paralleled the original European Council Directive 92/7/ECC (European Council, 1996) to determine suspension parameters. It is the step-down test, also known as the step test. This is a method where the vehicle is loaded and driven at low speed over an 80mm step. The resultant forces to the axle and chassis cause transient oscillations that may then be analysed for body-bounce frequency and damping ratio.

Gyenes *et al.*, (1992) suggested, under the broad term "parametric testing", driving a heavy vehicle slowly off a step with transient oscillations recorded using force transducers in the surface under the wheels. From that action, it was proposed that the resultant data be analysed for body-bounce frequency and damping ratio.

Cole & Cebon (1991) and Cebon (1999), amongst others, had reservations that the body-bounce frequency and damping ratio could be used to determine road-damaging potential. That research showed that conventional leaf-spring suspensions would pass a 2.0Hz criterion test for a step height of 80mm but fail for a 20mm step (Cole & Cebon, 1991). This due to the non-linear characteristics of leaf springs. Even so, Cebon (1999) endorsed the "step-down" test as a pragmatic approach to type approval and in-service inspections, based on the EC test procedures and specifications (European Council, 1996).

Peters (2003) derived suspension parameters from data recorded by an accelerometer attached to a semi-trailer during tests where all wheels on the test vehicle's tri-axle group were driven off 80mm high blocks simultaneously. This replicated the VSB 11 test to the greatest possible extent in a field situation. MATLAB™ software algorithms were applied to the recorded transient responses. The natural body-bounce frequency and damping ratio were derived using FFT and time domain analysis respectively.

Davis *et al.*, (2007) used a similar technique for the tri-axle group on a semi-trailer, the drive axle of a bus and a tag/drive axle group on a coach, an example of the latter is shown in Figure 6. That study noted that derivation of damping ratio was straightforward when all axles in a group are similar. It also noted that it became more difficult to determine damping ratio when two axles of different configuration were conjoined in the same group. The non-linearity of the shock absorber characteristic (Uffelmann & Walter, 1994) contributed to the anomalous results for damping ratios of different axles in the same group.

Figure 6. Step-down test apparatus used by Davis *et al*., (2007). Coach shown here.

Movement and excitation in one direction only would, however, be suitable if used in conjunction with data from an initial type-approval process similar to EC testing (Prem *et al*., 1998). Further, both Davis *et al*., (2007) and Sweatman *et al*., (1994) noted that, for a step-down forcing function, some mixed-mode signals were evident in the suspension response time series due to other fundamental vibration frequencies adding to the desired signal.

Simulation of this technique by computer-based analysis and small-scale experimentation indicated good agreement between suspension displacement, sprung mass acceleration and dynamic wheel-forces but that damping ratio was more problematic to determine accurately (Woodroofe, 1995). That this issue could be resolved by eliminating (or reducing to the greatest possible extent), inter-axle interaction by ensuring that all axles are tested in-phase was addressed in that work. That all axles could be dropped simultaneously to determine damper performance was proposed and validated, provided any articulation between (say) prime-mover and trailer was eliminated (Woodroofe, 1996; Woodroofe *et al*., 1988).

Derivation of parameters from excursions during small step-down perturbations is very effective because small excursions do not excite the bodies under test into non-linear behaviour (Prem *et al*., 1998).

3.1.4 Impulse excitation

Forsén (1999) used an approximation of an impulse as an input (forcing function) to a HV to excite vehicle vibration modes. By measuring the response of various vehicle components, he derived component damping ratios, fundamental frequencies and vehicle dynamic forces. The method used by Forsén to simulate the truck's suspension was to drive the HV over a 50mm thick plank 200mm wide at highway speeds. Excellent time-domain repeatability for this quasi-impulse input function was reported (Forsén, 1999).

Mitchell and Gyenes (1989) conducted tests including driving trucks over a 25mm plank placed on a test track of "medium roughness". By normalising the data from on-board instrumentation, the different suspension types were compared for natural body-bounce frequency and axle hop.

Davis & Sack (2004; 2006) and Davis (2005b) documented a low-cost process for testing heavy vehicle suspensions for in-service testing of RFS. This comprised a method where a truck was driven over a 50mm diameter pipe to simulate the equivalent of an impulse input function to a new RFS suspension, one axle at a time. Transient oscillations from the high-pressure air line to the air bags were recorded and analysed to determine the damped natural frequency and the damping ratio of the test vehicle's body bounce. Derived values were in good agreement with the manufacturer's certified values. A more expansive series of tests validated the previous testing and resulted in measured values for the fundamental frequency within 14% and for the damping ratio within 12% compared to the manufacturer's certification (Davis & Sack, 2006).

Sweatman *et al.,* (1994) tested HV suspensions for their characteristics using a step down after a ramp. This reference is noted under this section and not included above as a step-down test since the ramps and steady-state length before the step down used for those tests were shorter than defined by VSB 11 and the European Council Directive 92/7/ECC. They were closer to an impulse function in that the vehicle's suspensions were subjected to an impulse upwards and then downwards in quick succession. Good results, aligning with manufacturer's specifications were obtained. That report noted, as did Davis *et al.*, (2007), for multi-axle groups, test speeds need to be such that the suspension of one axle is allowed to recover to as near to quiescent or steady-state as possible before subjecting the following axle to the forcing function. Sweatman *et al.*, (1994) proposed $1ms^{-1}$ as an appropriate test speed. A speed of $1ms^{-1}$ was validated in the work by Davis & Sack (2006) and Davis *et al.*, (2007) noted that the limiting upper bound for valid results with recovery of suspensions after the impulse was approximately $2ms^{-1}$. These findings align with de Pont's (1999) validation of the concept of testing individual suspensions as well as whole vehicles.

3.1.5 On-road "white noise" excitation

Road excitation covers a range of frequencies but is generally not uniform in magnitude. However, Sweatman (1983) showed that, since there was interest only in a particular range of frequencies and their associated peaks in the frequency spectrum, the non-uniformity of the magnitude of the signal at any given frequency was therefore not significant. The 1983 report by Sweatman documented the design and implementation of a series of uneven-height additions to the drum of a roller-dynamometer. This effectively created a pseudo-random binary sequence simulating the frequency profile of a typical road as an input to a HV suspension. That effect was used to calibrate the instrumentation of test vehicles (Sweatman, 1983). A similar approach was taken in the rail operations area (Tanimoto, Nakata, & Yamamoto, 1989) with the use of a dynamometer to determine dynamic stability criteria for various types of rail suspension systems.

Hahn (1987a; 1987b) used on-road excitation of HV suspensions and performed testing over "good", "average" and "poor" roads to determine dynamic pavement loadings and, *inter alia*, deduced, for the test vehicle, natural body-bounce frequency and axle hop.

Davis and Sack (Davis & Sack, 2004) subjected a new RFS trailer to typical road travel. This excitation as an input signal to the test vehicle's suspension was assumed to approximate to a random or 'white noise' signal that contained all frequencies of interest as noted by Chesmond (1982). Analysis of the signals on the air springs enabled the natural body-bounce frequency to be derived with good accuracy with respect to manufacturer's specifications. Further testing in 2005 under an expanded test programme (Davis & Sack, 2006) validated the previous test hypothesis. Measured values for the fundamental body-bounce frequency were within 6% of the manufacturer's certified value.

3.1.6 Sinusoidal excitation

One of the proposed methods Gyenes *et al.,* (1992) summarised for determining axle hop and damped fundamental frequency of body-bounce was by performing a low-amplitude (1mm) sinusoidal sweep excitation (frequency scan) with all axles in phase on a road simulator. Road simulators are also known as "shaker beds" or "actuator test rigs" (Woodroofe, 1996). It was postulated that the eigenfrequencies of the sprung (body) and unsprung (axle hop) components could be found by the highest amplitude wheel-forces measured after such a sweep.

Sinusoidal inputs to test HV suspensions using simulators have been well documented (Hoogvelt, van Asseldonk, & Henny, 2004; Woodroofe, 1996). Prem *et al.*, concluded that constant amplitude sinusoidal sweeps and increasing-force frequency sweeps were of use in characterising suspensions for road-friendliness and in-service testing provided the latter were used in conjunction with type-test data (Prem *et al.*, 1998). Ahmadian (2003) noted that using this method on complete HVs in a reaction frame allowed the resonant frequencies of individual HV components to be found, particularly suspension components and the rails of the chassis.

3.1.7 Shock absorber testing

As noted above, shock absorbers are the key to RFS suspension health (Costanzi & Cebon, 2005). A direct method of testing suspension health is therefore to test the shock absorbers. This can be done using a dynamometer. Shock absorber dynamometers cycle a damper under test at defined speeds (usually involving a sinusoidal displacement of approximately 80mm at 1.3Hz). The forces resulting from the resistance of the damper during this process are measured. Forces and velocities are then plotted for force/displacement and force/velocity and compared with manufacturer's recommended metrics for serviceability. The shock absorber dynamometer reported in the Sweatman, *et al.*, (2000) study was found to have a repeatability error of approximately 2%.

3.2 Discretionary test inputs and associated equipment

3.2.1 Servo-hydraulic actuator test rig excitation

Servo-hydraulic actuators are used to excite the suspensions of heavy vehicles and are built into test rigs described as "shaker beds" or "road simulators". The vehicle under test has its wheels supported on plates connected to hydraulic actuators. These actuators are made to vibrate via electronic control of their associated hydraulic valves. By varying the inputs to the drivers of the actuators, different input functions may be applied to the wheels of the vehicle under test. By measuring various dynamic forces either on the vehicle or at the plates under the wheels, HV dynamic parameters may be derived. It takes considerable effort to set up the input signal data for any given type of test. Reliable test conditions may be replicated and, if only dynamic wheel forces are required, they do not require the vehicle under test to be instrumented, just the support plates for the wheels. Accordingly, many vehicles may be tested for dynamic wheel forces (de Pont, 1992).

The disadvantage with these rigs is that, when sufficient actuators (12 for a 6-axle semi) are provided to excite each wheel of the test vehicle, they are expensive (de Pont, 1997). Such expense is not such an issue for (say) HV manufacturers or research facilities who use these machines of necessity (Hu, 1988). Even so, when many tests are performed costs per test are cheaper and more convenient than installing instruments on individual vehicles (de Pont, 1992).

Axles (whole-of-vehicle) being dropped simultaneously using a shaker bed to determine damper performance was validated (Woodroofe *et al*., 1988), provided any articulation between (say) prime-mover and trailer was eliminated with simultaneous inputs at all wheels.

de Pont (1992; 1997) used instrumented vehicles in an attempt to develop cheap and reliable assessment procedures to determine the "road friendliness" of any given heavy vehicle suspension. The test HVs were instrumented to record dynamic loads on various components as they were driven on typical roads. Using the recorded data as inputs to the controls on the hydraulic actuators, a 2-post servo-hydraulic tester was programmed to replicate the forces measured during the road tests and apply them to the test vehicles.

An instrumented vehicle to calibrate a weigh-in-motion (WiM) installation was described by Hoogvelt *et al*., (2004). The vehicle was instrumented and on-road data were recorded. In the process, the vehicle's suspension was calibrated using a shaker bed and the dynamic characteristics derived. Other researchers have calibrated the instrumented suspensions of test HVs with shaker facilities (LeBlanc & Woodrooffe, 1995).

Cebon described the construction of a "quarter-car" test rig which used hydraulic actuators to impart wheel forces (Cebon, 1999). Using various inputs to the hydraulic actuator drive system, inputs such as "white noise", a "step-up" and a "step-down" were used to determine the parameters of the suspension under test. This work showed that the step-up input should be followed by the step-down to obtain accurate damper parameters. This aligned well with the concept of a "bump test" explored by Davis *et al.*, (2007; 2004; 2006) and Sweatman *et al.*, (1994).

That these machines would be suitable as one means of type-testing HV suspensions for "road-friendliness" was noted by Prem *et al*., (1998). Sweatman and Addis (1998) also noted that these machines were valid for replication of dynamic wheel-loads. That study went on to state that if DLC were the chosen measure for dynamic loadings, 97% accuracies could be achieved by their use.

Stanzel & Preston-Thomas (2000) reviewed the use of these facilities regarding whether accurate test input signals could be derived for these test beds. That study concluded that more development of test signal envelopes was needed but noted that test signals recorded from on-road testing were generally valid as actuator input signals provided all axles were excited.

3.2.2 Stationary reaction frame

This is a device where the HV or part of a HV (e.g. axle and wheels) to be tested is attached rigidly to a frame bolted to the solid floor of a test facility. The HV's hubs and other components desired to be measured are instrumented. Test signals are then applied to the HV's hubs using hydraulic actuators controlled electronically with test signals similar to those used in shaker bed testing. Ahmadian *et al*., fixed the chassis of a test HV to the floor via stiff frames and applied this technique to test the roll stiffness, vertical stiffness and roll-steer of HV suspensions (Ahmadian & Ahn, 2003).

A variation on this theme was a test rig comprising a cast iron table with the necessary parts of a surrogate HV suspension mounted to it. This to simulate sufficiently a HV suspension. Optimum values of damping ratio and spring constant for minimising road damage were derived by the application of programmable forces from hydraulic actuators (Cole & Cebon, 1995).

O'Connor *et al*., (O'Connor, Kunjamboo, & Nilsson, 1980) used a stationary reaction frame to test a part-axle of a HV in an attempt to determine the spring constants for dual tyres and steel leaf springs of a test HV. The use of this type of servo-hydraulic tester was explored in a separate paper by Ahmadian (2003) who noted its flexibility in applying "hard bump" input signals which simulated the HV encountering a pothole and field-measured road signals. As with all simulations, the *caveat* applied (Ahmadian, 2003) in that these machines should be used in conjunction with field testing since they can't replicate all real-world situations. One positive benefit noted was better repeatability when compared with field testing (Ahmadian, 2003).

4 Evaluation of HV Suspensions – on-vehicle instrumentation

4.1 General

The following two sections are deliberately separate from the forcing function descriptions in the preceding section. This because the instrumentation is not necessarily dependant on the test method and *vice versa*.

Measurements by transducers mounted on the HV are used to derive forces on the unsprung (axles & wheels) or sprung (chassis) parts of a heavy vehicle. These are detailed in this section. The derivation of these forces is for various reasons as discussed later.

4.2 Wheel hub transducers

These expensive devices use strain gauges, and sometimes accelerometers, in a hub adaptor to measure continuous wheel-force data generated directly from the wheels of the heavy vehicle under test (Forsén, 1999; Hahn, 1987a; Jarvis & Sweatman, 1982; Sweatman, 1983; Whittemore, Wiley, Schultz, & Pollock, 1970). General Motors first developed them. Where accelerometers are included, they are used to correct for the inertial effects of the masses outboard of the strain gauges, usually the wheel and tyre (Cebon, 1999). In that way, the operation of these devices to derive wheel forces is similar in theory to the combination of strain gauges and accelerometers described below and for the same reasons. Wheel hub transducers may also be configured to measure lateral wheel forces (Gyenes *et al.,* 1992).

Such transducers can be used to measure forces on the heavy vehicle as it is being driven on roads or subjected to simulated road-induced dynamic loads in a dynamometer or road simulator. For Sweatman's research (1983) only one hub was instrumented per vehicle due to the cost. Forsén (1999) developed parametric models for a 2-axle prime-mover by recording vehicle dynamics during on-track testing. Computer simulation models were developed and compared with the recorded data. This was for purposes of determining ride comfort, chassis modal vibration, wheel forces and sprung mass accelerations. That study used two wheel-hub transducers on the front axle to measure front axle forces and torque as well as other transducers covered below. The development of these measurement devices has continued with sophisticated software to process the signals now integrated into the measurement package (Hachmeister, 2000).

4.3 Strain gauges

4.3.1 General

Strain gauges fixed to the axles of a HV may be used to find, variously, the bending moment, shear force or strain on a particular part of the HV during dynamic or static loading. These data may then be used to derive dynamic wheel-forces, body or axle loads or other dynamic behaviour (de Pont, 1997). Mitchell and Gyenes (1989) amongst others (Gyenes *et al.*, 1992; Gyenes & Simmons, 1994), tested dynamic parameters of different HV suspension types (*viz*, rubber, air and steel sprung). Wheel forces were derived in these tests and a contribution to their measurement was from strain gauges mounted on the axle between the hub and the spring.

de Pont (de Pont, 1992; 1997) recorded on-road forces from various test HVs during test programmes to develop cheap and reliable “road-friendliness” procedures to be applied to a wide range of HVs. Amongst other transducers (the output of which made a complete reconstruction of the wheel-forces possible) the test vehicles had strain gauges fixed on each half-axle between the spring and the hub to measure the dynamic bending and shear forces. These transducers were acknowledged as the simplest method for determining vertical dynamic tyre forces (Cebon, 1999). Mounting points vary depending on the force measured.

4.3.2 Strain gauges to measure axle bending moment

Cebon (1999) noted that many researchers (Mitchell, 1987; Mitchell & Gyenes, 1989; Woodroofe, LeBlanc, & LePiane, 1986) have measured the bending moment on the axles by mounting the strain gauges on the top and bottom surfaces of the axle between the spring and the hub. Wheel forces proportional to the signals from these gauges may then be determined if the necessary correction for the mass outboard of the strain gauge is made. LeBlanc *et al*., (1992) noted however, that mounting strain gauges on the top and the bottom of the axle had the disadvantage of introducing inaccuracies. These come from adding:

- lateral tyre forces;
- vehicle body-roll forces; &/or
- forces due to changes in the length of the moment arm distance (distance 'x' in Figure 9) from axle roll

to the axle bending moment signal.

LeBlanc *et al*., (1992) recommended that strain gauges be mounted on the sides of the axle to eliminate these extraneous signals which are difficult to remove and for which it is difficult to compensate. Other research has noted that use of bending gauges created unsatisfactory correlation between on-board instrumentation and wheel forces measured at the tyre/ground interface (Stanzel & Preston-Thomas, 2000). An advantage of bending strain gauges (top and bottom of axle) is that bending moment is larger than shear force at the same lateral point on an axle, resulting in large signals to noise ratios. The bending moment signals are approximately 2.5 times greater than those for axle shear force measured by the same strain gauge (LeBlanc *et al*., 1992).

4.3.3 Strain gauges to measure axle shear force

In-field testing has shown in the recent past that the technique of side-mounted strain gauges to measure axle shear forces results in slightly reduced, but still usable, signal levels (de Pont, 1997; Jacob & Dolcemascolo, 1998) without the added complication of lateral forces being included in the data. The signal size disadvantage of side-mounting of strain gauges was either accepted and the signals used regardless (de Pont, 1997) or has been accommodated by modification (Hoogvelt *et al.,* 2004). A mechanical strain amplifier was developed by Hoogvelt *et al.,* (2004) that:

- used two thin bars that had an order of magnitude smaller stiffness; and

- were configured in an "X" formation across the horizontal neutral axis of the axles of a test HV.

The reduced sections of the bars increased the amount of strain measured. Sweatman *et al*., (1994) used strain gauges to determine shear forces, rather than bending moments, at the axle ends.

Whichever mounting point is chosen, data from axle-mounted strain gauges needs to be combined with accelerometer data (see below) if accurate wheel-force data are to be derived.

4.3.4 Strain gauges to measure wheel forces indirectly from tyre deflection

Gyenes *et al*., (1992) proposed that tyre sidewall bulge could be measured with strain-gauged cantilever arms. That report concluded that distortion of tyre sidewalls:

- was proportional to tyre-force within 5%; and
- could be used as an option for mass testing of parameters for HV suspension health up to axle-hop frequencies of 20Hz.

No experiments arose from this suggestion.

Chang *et al*., (1998) attempted to calibrate tyre deflection to measure HV wheel forces using a capacitance displacement transducer system. The reference signals for that testing were provided by strain gauges mounted on the HV axle. That paper was not definitive regarding the positioning of the strain gauges: the issue of compensating for or removing the effects of lateral wheel-forces from the derived wheel-force data was not addressed.

4.4 Accelerometers

4.4.1 General

Accelerometers fixed to the chassis of a HV can be used to derive body-bounce and damping ratio parameters with good experimental accuracy (Peters, 2003). When fitted to the axles of a HV, accelerometers can be used to determine axle-hop frequencies or wheel-hop frequencies if mounted close to the hub. Forsén (1999) used 16 accelerometers on a 2-axle prime-mover to measure 3-axis accelerations at the front axle, cabin, front and middle of the chassis and vertical acceleration at the rear axle to generate whole-of-body measures of acceleration.

4.4.2 Accelerometers mounted on axles

Strain gauges mounted on the axle of a HV, as covered in the previous section, contribute to the reconstruction of HV wheel-forces. In addition to strain gauges, accelerometers are necessary to measure the dynamic inertial loads imposed by the unsprung masses outboard of the strain gauges and therefore complete the HV wheel-force history.

Gyenes *et al*., (Gyenes *et al.*, 1992; Gyenes & Simmons, 1994) and Mitchell and Gyenes (1989) installed an accelerometer on each HV half-axle between the strain gauge and the hub. These were used to determine inertial forces from the unsprung masses outboard of the axle-mounted strain gauges, a technique used also by others (Cebon, 1999; Hoogvelt *et al.*, 2004; LeBlanc *et al.*, 1992; Woodroofe *et al.*, 1986). Accordingly, correction was applied to determine the contribution made to the overall wheel-force loads by the dynamic inertial loads of these masses.

Sweatman *et al*., (1994) used accelerometers to determine any unsprung mass inertial forces outboard of axle strain gauges. Using the technique described in Section 4.5, shear forces rather than bending moments were measured at the axle ends to determine total wheel forces in that work.

Huhtala & Halonen (2002) used the derived wheel forces from an instrumented vehicle to calibrate a Weigh-in-Motion (WiM) installation. The vehicle was instrumented with accelerometers and strain gauges on the axles in the same way as Gyenes *et al*., (1992) and Mitchell and Gyenes (1989).

Accelerometers as close as possible to hubs of a test HV were used by de Pont (1992; 1997) to contribute to building a complete profile of wheel forces from on-road tests so that these could be used as inputs to servo hydraulic excitation.

4.4.3 Accelerometers mounted on the body of a HV

Accelerometers were described by Ahmadian (2003) for measuring:

- horizontal &/or vertical accelerations at the front, middle and rear of a HV chassis;
- above the driver in the cabin;
- behind the driver on the rear wall of the cabin; and
- above the front drive axle (of a tandem-drive prime-mover)

during servo-hydraulic rig testing.

de Pont’s (1997) recording of on-road forces from a 3 axle tanker dog trailer also used accelerometers mounted on the chassis above each wheel and accelerometers mounted elsewhere on the chassis to measure sprung and unsprung mass behaviour.

In addition to wheel-force measurements, Sweatman *et al*., (1994) and Gyenes and Simmons (1994) amongst others, used accelerometers mounted on the chassis of a test HV to measure sprung mass movements during testing.

Hahn (1987a; 1987b) performed testing by driving test HVs over ”good” “medium and “poor” road sections. The instrumentation for these tests consisted of accelerometers mounted directly in the hubs and on the chassis of the HVs. This obviated the necessity for correction due to unsprung mass acceleration outboard of any axle strain gauges. By knowing the respective masses of sprung and unsprung components of the vehicles, the inertias were determined. Multiplying the respective inertias by the accelerations for sprung and unsprung masses allowed determination of wheel forces. This technique is only suitable for simple vehicles such as buses but not for semi-trailers, for instance (Cebon, 1999). Even so, errors may be introduced from the different vibrational modes of the chassis or load distribution of axle groups redistributing sprung axle mass (Cebon, 1999; de Pont, 1992, 1997; Hahn, 1987a).

4.5 Accelerometers + strain gauges = wheel forces

Accelerometer signals added to signals from strain gauges may be used to derive wheel-force measurements of HVs. The dynamic wheel-force F_{wheel} can be expressed by the formula (Woodroofe *et al.*, 1986):

$$F_{wheel} = F_{shear} + ma$$

Equation 13

Where:

a = acceleration of the mass outboard of the strain gauges;

m = mass outboard of the strain gauge; and

F_{shear} = shear force at the strain gauge on the axle-end of interest.

The value of m, representing the unsprung masses outboard of the strain gauges, needs to be determined and acceleration a measured simultaneously with F_{shear} data.

A set of strain gauges mounted on the half-axle between the hub and the spring will provide a signal proportional to the shear force at that point on the axle. The forces due to the unsprung mass outboard of the strain gauges contribute to an inertial force component from the movement of those masses located between the strain gauge and the tyre contact patch. These forces may be derived by multiplying the value of the unsprung mass outboard of the strain gauges by the acceleration as measured by the accelerometers at the hub. Adding the shear force value and the acceleration signal, after applying appropriate calibration factors, yields a surrogate for the dynamic instantaneous vertical force imparted to the pavement by the wheel of a HV (de Pont, 1992, 1997; Gyenes *et al.*, 1992; Gyenes & Simmons, 1994; Hoogvelt *et al.*, 2004;

Mitchell & Gyenes, 1989; Simmons & Wood, 1990; Woodroofe *et al*., 1986). Appendix 1 shows the derivation of Equation 13. Appendix 5 shows the method for calibrating the strain gauges (Woodroofe *et al*., 1986).

Huhtala & Halonen (2002) and Sweatman *et al*., (Sweatman *et al*., 1994) used vehicles instrumented with accelerometers and strain gauges on the axles in the same way as did Gyenes *et al*., (1992) and Mitchell and Gyenes (1989) to re-construct dynamic wheel-forces.

Claimed accuracies were 95-97% using this approach (Gyenes & Simmons, 1994; Jacob & Dolcemascolo, 1998; Mitchell & Gyenes, 1989). Later work concluded that this could be improved to 98-99% (Gyenes & Mitchell, 1996; Hoogvelt *et al.,* 2004).

4.6 Air-spring pressure transducers

Installed in air lines connected to the air bags of a RFS-equipped vehicle, air pressure transducers yield data that is a surrogate of the pressure in the air bags. The pressure in the air bags of a HV is proportional to the mass supported by the air bags (Davis & Sack, 2004). This reading, plus an allowance for the unsprung mass, is used by HV on-board mass measurement systems to determine the gross vehicle mass (GVM) of HVs (Transport Certification Australia Limited, 2007). Static testing has shown that there are systems available which are accurate to better than 98.5% (Transport Certification Australia Limited, 2007). Provided the on-board system or instrumentation recorder is fast and accurate enough, both static and dynamic data may be derived when the vehicle is stationary or travelling respectively.

Testing by Davis and Sack (2004; 2006) used FFT analysis of air-spring pressure from a RFS trailer during typical road travel to drive the natural frequency of the trailer's body bounce.

4.7 Displacement transducers

Displacement transducers include such instrumentation as string potentiometers and linear variable displacement transducers (LVDTs). These have been used for measuring the dynamic variations in the distance between the chassis and the axle of a HV.

Gyenes and Simmons (1994) used displacement transducers to determine axle-to-body movement due to spring flexure under test conditions, as did Sweatman *et al*., (1994) and others (Dickerson & Mace, 1981). Forsén (1999) used displacement transducers to measure vertical axle displacements as well as the longitudinal displacement between the engine/gearbox and the chassis.

The physical displacement between the chassis and body of a HV may be used to derive suspension parameters and determine RFS health (Australia Department of Transport and Regional Services, 2004a). Displacement transducers such as LVDTs and string potentiometers are used for VSB 11 testing to measure the transient body-to-axle distances during the specified tests. Fixing the string potentiometer between the axle and the chassis is one method listed in the ARTSA air suspension code of practice (Australian Road Transport Suppliers Association, 2001).

The tests devised by Milliken *et al.*, (2001) to determine damping ratio and natural body-bounce frequency nominated string potentiometers as measurement devices. They were attached to the chassis and measured the change in relative displacement between the sprung and unsprung mass.

LVDTs were described by Ahmadian (2003) for measuring velocity and displacement of a HV cabin during servo-hydraulic rig testing. That paper also described their use for displacement measurements between the front, middle and rear of the chassis and the ground.

de Pont (1997) used a linear variable differential transformer (LVDT) to measure dynamic variations in suspension deflection between the axles and the chassis of a test vehicle during on-road excursions. These signals were used as inputs to generate servo hydraulic excitation signals.

4.8 Tyre pressure transducers

Bridgestone (2004) announced some time ago that a tyre-pressure monitoring system would be developed which used a sensor in the tyre to measure air pressure and temperature. By transmitting these data to a receiver on the underbody of the vehicle, tyre pressures could be monitored. The sampling rate for the data was nominated (Bridgestone, 2005) as one sample every 7 seconds, i.e. a frequency of 0.14Hz which would be too low to be of use in determining HV body-bounce characteristics and hence RFS health.

Apart from this potential application for measuring body-bounce, other researchers (Addis, Halliday, & Mitchell, 1986; Dickerson & Mace, 1981) have used tyre pressure as an indirect indicator of wheel forces. Some have been successful, some not. Tyres have a non-linear force/pressure relationship response characteristic which is dependant on speed so care needs to be taken to calibrate at the particular speed(s) of the tests (Popov, Cole, Cebon, & Winkler, 1999). Hu did derive the force/displacement relationship for HV tyres, but that was in a static test situation on a shaker bed and the wheels were not rolling (Hu, 1988). The OECD concluded that pressure measurement to derive vehicle wheel loads was a relatively unsuccessful method (OECD, 1992).

Work in the 1990s indicated that, depending on speed, the inaccuracy inherent in measuring wheel-force via tyre pressure varied -25% / +200% (Cebon, 1999). Further to this Popov, *et al.*, (Popov *et al*., 1999) showed that:

- the rolling resistance and frequency transfer function of a typical truck tyre vary with speed; and
- the stiffness and damping characteristics of a tyre varies with speed.

This makes it difficult to measure tyre pressure with any accuracy or to correlate tyre dimensions to wheel load. A tyre calibrated when stationary will not be able to be measured to derive useful data when moving (Cebon, 1999). The corollary of this is that the tyre must be rolling when it is calibrated (LeBlanc *et al.*, 1992) although given the findings of Popov, etc., it is difficult to determine at which speed this should be done for reliability of measurement over the pragmatic issue of variation in speed that will occur during any test.

The difficulty in correlating static and dynamic tyre stiffnesses was also was noted from tests with shaker beds (Stanzel & Preston-Thomas, 2000).

4.9 Laser detectors

Magnusson (1987) used lasers to measure the distance between the axle and the ground for a bus and attempted to infer wheel loads due to tyre compression. Accuracy was within 92% and dependant on speed.

ARRB project AT1212 is attempting to use a similar arrangement to determine wheel loads for different HV suspensions (George & Blanksby, 2006).

Lasers were used in the Transport Research Laboratory (TRL) in the UK by Dickerson & Mace to measure axle-to-road displacement and derive wheel-forces therefrom (Gyenes & Simmons, 1994). This testing (Dickerson & Mace, 1981) determined wheel loads indirectly from the vertical deflection of the rear tyres of a 2-axle HV. These tests used lasers measuring the distance between the axle and the road surface at two different speeds. Mace and Stephenson (1989) instrumented a two-axle semi-trailer with a similar arrangement of lasers measuring the distance between the axle and the road surface to measure tyre deflection of a semi-trailer and quantified the dynamic variation in wheel-load. Both these test programmes were successful due to:

- many runs at constant speeds;
- averaging the unevenness of the road surface over relatively long data recording runs;
- rigorous pre-test calibration; and
- determination of tyre force/load/deflection characteristics at the selected speeds before testing.

Lasers mounted on frames behind the front and rear axles of a two-axle truck have been used to try to determine road profiles (Arnberg, Holen, & Magnusson, 1992). That work examined the effect that axle loads had on the road profile at the surface but did not draw conclusions about whether wheel forces could be derived by measuring the distance between the body of a HV and the ground.

4.10 Infrared sensors to determine variation in mass indirectly from tyre deflection

Cantieni (1992) used infrared distance sensors mounted under the axles and on the hubs of test HVs to measure deflection of HV tyres and derived wheel loads directly from the vertical deflection of the tyres at various speeds. That study noted, p69 (1992):

> "Since the spring characteristic of a tyre is not linear, the conversion of the voltage signal delivered by the measurement system into Kilo-Newtons was quite involved."

The results were within acceptable experimental error.

4.11 Capacitance sensors to determine variation in mass indirectly from wheel rim flexure

The variation in wheel-rim distortion due to load was measured with a calibrated capacitance displacement sensor to determine wheel-forces (Chang *et al.*, 1998). The change in displacement between an arm cantilevered off the axle and the rim of the wheel of the HV was measured by variation in capacitance. Whilst not explored in detail in the paper, that system presumably relied on flexure of the rim of the wheel and assumed that this was proportional to wheel load. Eccentricity of wheels and tyres led to a 20% error in this method.

4.12 Direct measurement of suspension damper health

Sweatman, McFarlane, Komadina, & Cebon (2000) included a reference to a “shock absorber integrity device”, the function of which was to sense significant alterations to tyre inflation pressures or shock absorber wear. D Cebon (Cambridge University), RJ Hammond and RJ Dorling lodged a patent application for a “suspension integrity monitor” in 1996. The patent was not granted since the application was withdrawn before grant (Tilbury, 2005).

4.13 Infrared test to determine shock absorber function

Shock absorbers, by design, get hot when they are working. Heat generated by movement of hydraulic oil through the internal mechanism of the damper is dissipated by the damper body. If a shock absorber is working, infrared radiation should be seen by the appropriate detectors. This may not be an accurate approach but it could be useful for a pass/fail test.

The Roads and Traffic Authority (RTA) of NSW uses infrared (IR) sensing units, so-called ‘heat guns’, to perform roadside checks for functional *vs.* non-functional brakes on heavy vehicles but this is in the context of an in-service assurance of roadworthiness. No current uses of heat detection to determine damper functionality were found to have been implemented.

ARTSA (Australian Road Transport Suppliers Association, 2001) cited a method where a “heat gun” (hand-held thermometer) could be used to determine if shock absorbers are functioning. No judgement or guidance was given regarding the accuracy of this method.

Blanksby *et al*., reported that some shock absorbers temperatures recorded were at or near ambient (2006) after operation. This may have indicated that they were inoperative or just fitted to a HV operating on pavements in good condition. Many trailer suspensions have their shock absorbers mounted almost horizontally to reduce suspension envelope. This results in short damper stroke during normal operation;

axle movements being compensated by compression of the bushings with little or no heating in the shock absorber.

Davis noted a similar phenomenon during the 2005 testing of a quad-axle semi-trailer (2005a) with conventional shock absorber mountings and travel over “good” pavements.

4.14 Summary of this section

The following table is a broad summary of the features of the instrumentation described in this section for deriving on-board measurements of suspension parameters. Distance or displacement is denoted here as the time-dependant variable *x*.

Table 1. Summary of instrumentation features for on-board measurement of HV parameters

Instrumentation	Position	Measurement	Accuracy	Complexity		Capital cost per unit	Data post-processing
				installation	calibration		
Wheel hub transducer	on/in hub of HV	strain & acceleration[1]. lateral and vertical wheel forces	very high	high[2]	High	high	high
Strain gauges – bending moment	top &/or bottom of axle between the hub and the spring	strain. vertical and lateral (undesirable) wheel forces[3]	high	moderate	moderate	low	moderate
Strain gauges – shear force	side of axle between the hub and the spring	strain. vertical wheel forces[3]	high	moderate	moderate	low	moderate

[1] optional but desirable
[2] requires wheel-hub adaptor
[3] when the signals are combined with accelerometer signals measuring the dynamic inertial forces from the unsprung mass outboard of the axle strain gauges

Instrumentation	Position	Measurement	Accuracy	Complexity		Capital cost per unit	Data post-processing
				installation	calibration		
Strain gauges – tyre deflection	cantilever arms onto tyre sides	vertical wheel forces	never tested but probably low	never tested but probably high	never tested but probably very high	low	never tested but probably moderate
Accelerometer	chassis	$\ddot{x}$. body-bounce, damping ratio[4] & chassis acceleration[5]	high	low	Low	high	moderate
	axle	$\ddot{x}$. axle-hop[4], wheel forces[6]	high	low	Low	high	moderate
	hub	$\ddot{x}$. wheel forces[7]	high	low	Low	high	moderate
	cabin	$\ddot{x}$ harshness, cabin vibration	high	low	Low	high	moderate
Air pressure transducer	air springs or air line	forces between chassis and axle	Very high	low	High	moderate	low

[4] by double-integration
[5] when combined with the accelerometer signals from the axle, allows wheel forces to be derived
[6] when combined with the accelerometer signals from the body/chassis, allows wheel forces to be derived
[7] when the signals are combined with strain gauge signals measuring the forces from the HV sprung mass transmitted from the springs to the axles

Instrumentation	Position	Measurement	Accuracy	Complexity		Capital cost per unit	Data post-processing
				installation	calibration		
LVDT	connected between chassis and axle	*x*. forces between chassis and axle	very high	low	moderate	high	low
String potentiometer	connected between chassis and axle	*x*. forces between chassis and axle	moderate	very low	moderate	low	low
Laser detector	axle or hub	*x*. displacement proportional to wheel forces via tyre deflection	high[8]	high	High	high	moderate
Infrared	axle or hub	*x*. displacement proportional to wheel forces via tyre deflection	moderate[8]	high	High	high	moderate
Capacitive displacement	cantilevered arm off axle	*x*. displacement proportional to wheel forces via wheel deflection	low	high	High	moderate	moderate
Infrared	hand-held	radiated heat	moderate	low	moderate	moderate	low

[8] provided test speeds and pavements are constant per test since tyre deflection varies non-linearly with speed, tyre pressure, road roughness, etc.

5 Evaluation of HV Suspensions – off-vehicle wheel-force measurement

Measurements by transducers mounted on or in the pavement have been used to derive applied forces from heavy vehicles. These are detailed in this section.

5.1 Weigh in motion

Weigh-in-motion (WiM) comprises systems where wheel-force sensors are attached to, or buried in, the road surface or culverts. These may be either piezo or other types of weigh-in-motion (WiM) sensors such as strain gauges. Many trials have been performed such as those documented by Hahn (Hahn, 1987a) and the EC (2000) using multiple arrays of sensors in the road surface to detect and compare wheel forces. Provided sufficient numbers are installed, the truck's "suspension wavelength" may be reconstructed (Brannolte, Griesbach, Youssef, & Opitz, 2002; Gyenes *et al*., 1994; Potter *et al*., 1997).

Potter *et al*., (1997) concluded that comparing WiM signals to in-specification RFS damped natural body-bounce could be used for in-service screening of poorly maintained suspensions. Prem *et al*., concluded that arrays such as these could be used as a means of type-testing HV suspensions for "road-friendliness" (Prem *et al*., 1998).

The COST 323 project (Caprez, 1997) installed 6 different types of vehicle classification systems in a test pavement length. All but one (the PAT comprised bending plate technology) used either capacitive or piezo sensors. That report concluded that none of the systems tested could determine the static mass of the vehicles due to the dynamic components of their wheel forces. The authors of the report were not satisfied with that result because the aim was to find a system that could determine vehicle GVMs equivalent to static mass during travel. However, the report noted that the body-bounce "wavelength" could be measured using in-pavement sensors. A similar judgement was made on recordings of dynamic loads measured during the WAVE project testing (European Commission, 2000).

Brannolte *et al.,* (2002) analysed the results from the technologies for optimising WiM

(TOP) trial where 560 truck passes were made over a WiM array. That study was to validate algorithms for determining static wheel-forces from a WiM installation. Whilst the primary focus of the TOP research was not to determine HV dynamic parameters, the unexpected outcome was that vehicle body-bounce and axle-hop dynamics could be derived readily from the outputs of the in-road sensors. Accordingly, an array of WiM sensors may be used to re-construct heavy vehicle suspension wavelengths from dynamic WiM measurements (Brannolte *et al.*, 2002; Potter *et al.*, 1997).

One disadvantage reported consistently with regard to WiM is that it is dependant on sound, smooth, flat pavement leading up to the measurement site to achieve accurate measurement. In this context, "accurate" is used to mean "agreement between static and dynamic force measurement." Ironically, the effect of pavement perturbations excites the modes of vibration of interest in HVs and allows WiM measurement of HV suspension wavelengths.

Arrays of in-road sensors (WiM) could be used for screening purposes to detect out-of-specification suspensions which, combined with other detections methods could flag specific vehicles for further attention at WiM sites (Prem *et al*., 1998). This could include the use of cameras for identity detection (e.g. via automatic number plate detection) if penalties were to be imposed (Doupal, Calderara, & Jagau, 2002).

5.2 Capacitive mat

A WiM system using a mat of capacitive sensors incorporated into polyurethane tiles was proposed, developed and trialled extensively by Cole *et al*., (Cole & Cebon, 1989; Cole, Collop, Potter, & Cebon, 1992; Cole *et al*., 1996), Collop *et al*., (1994) and Potter *et al*., (1995). These tiles were placed over a 56.4m length of the A34 near Oxford (UK). Dynamic forces imparted by normal traffic were recorded and tyre forces from an instrumented vehicle were used to validate the measured forces from the mats. HV body-bounce and axle-hop frequencies could be seen clearly from the spectral density plots of the wheel forces.

Correlation between wheel-forces derived from an instrumented vehicle and the capacitive mats was reported to be good to very good (Potter *et al.*, 1995), although in later work, Potter *et al*., (1997) reported that at least 250m of mat length was required for sufficient repeatability for 95th%ile accuracy. Countering this, the DIVINE project final technical report, p67 (OECD, 1998) noted, regarding the A34 sensor array and testing:

"Some measurements made by the TRL in 1992/93 on Motorway A34, at Abingdon (United Kingdom), were also considered. The pavement conditions were rather poor with an uneven surface (unknown IRI value), rutting and some cracking. An array of 48 uniformly spaced half-capacitive strip sensors (24 lines) was used. Because of some sensor problems, many of the measurements have been considered to be unreliable and only a few results are reported here."

5.3 Strain gauges

5.3.1 General

As noted in Section 4.3, strain gauges are widely used for measurement of mechanical loads. These devices are particularly versatile, relatively cheap, durable, and have good-to-excellent accuracy and precision.

5.3.2 Bridge measurement

Strain gauges are used to determine dynamic bridge loadings and this technique has been widely reported in studies such as Ashebo *et al.,* (2007), Senthilvasan *et al.,* (Senthilvasan, Thambiratnam, & Brameld, 2002) and Schiebel *et al.*, (2002) to mention some. Usually this is in conjunction with other instrumentation such as linear displacement potentiometers &/or accelerometers as listed in Sections 5.4 & 5.5.

5.3.3 Strain gauges in instrumented pavements

Middleton & Rhodes (1994) instrumented a private haul road with 126 strain gauges under the base course. Data from these gauges was used to measure the frequency of dynamic wheel loads reliably. Dynamic load coefficients (DLCs) and dynamic loading values were derived from the measured wheel forces.

Hahn (1987a; 1987b), during testing of HVs for dynamic forces, validated on-board HV instrumentation against dynamic readings from an instrumented pavement which had 13 strain gauges installed at intervals of 2m. Similarly, Mitchell (1987) and Gyenes & Mitchell (1992) used strain gauges installed in pavements of varying test lengths to examine dynamic loads. That testing led to conclusions about spatial repeatability of HV wheel loads and differences in wavelengths for differing HV suspension types.

5.3.4 Strain gauges in vehicle testing machines

Sweatman *et al*., (1994) used the VicRoads electronic mass unit (EMU) to measure dynamic wheel forces under the wheels during step-down tests on HVs. The EMU, similar to other portable electronic enforcement weighing devices incorporates a load cell (Sweatman *et al*., 1994) which usually comprises a strain gauge or gauges.

Stationary reaction frames and shaker beds have strain gauges in the form of load cells under the wheel supports to measure dynamic wheel forces, as reported widely (de Pont, 1992, 1997; Hoogvelt *et al*., 2004; O'Connor *et al*., 1980), and usually as part of the calibration process for test HVs in larger studies.

Strain gauges were used by Ahmadian *et al.*, (2003) for measuring HV rear wheel forces at the actuators during servo-hydraulic rig testing.

5.4 Displacement transducers

“Spring and wire” displacement transducers (similar in operation to a string potentiometer) have been used to determine the dynamic deflections of bridges

(Austroads, 2003; Heywood, 1995). Hu (1988) used a string potentiometer to measure dynamic variations between the support plates of a road simulator (shaker bed) and the hub of a HV in an early attempt to derive dynamic wheel forces by measuring tyre deflection.

Dynamic deflections were measured from test bridges as HVs travelled over them in separate studies (Cantieni, 1992; Moldoveanu & Heywood, 1997; Senthilvasan *et al*., 2002). These used displacement transducers anchored to the bridge structure at one end and the river bed at the other to measure bridge deflection (Cantieni, 1992; Moldoveanu & Heywood, 1997).

Further, linear variable displacement transducers (LVDTs) have been used for measuring the dynamic variation in distance from bridge decks to the corresponding riverbed (Moldoveanu & Heywood, 1997).

LVDTs were described by Ahmadian (2003) for displacement measurements between the front, middle and rear of a test HV chassis and the ground.

5.5 Accelerometers

Some early research attempted to find less-damaging suspension types and proposed the use of tyre deflections to determine wheel-loads. One approach (Hu, 1988) used accelerometers on the wheel-support plates of a shaker bed to ensure that the same simulated road signal was applied each time to the wheels of the HV under test. This technique was also used to calibrate a test HV for static forces during wheel-load calibrations in a larger study of weigh-in-motion systems (Hoogvelt *et al.*, 2004).

Cantieni instrumented a bridge with accelerometers to measure the bridge eigenfrequencies as various test HVs were driven across it (Cantieni, 1992). Australian studies of bridge-vehicle interaction used similar instrumentation to determine vibration modes of structures similar to those found by Cantieni (Moldoveanu & Heywood, 1997; Senthilvasan *et al*., 2002).

5.6 Summary of this section

The following table is a broad summary of the features of the instrumentation described in this section for deriving off-vehicle measurements of suspension parameters. Distance or displacement is denoted here as the time-dependant variable x.

Table 2. Summary of instrumentation features for off-vehicle measurement of HV parameters

Instrumentation	Position	Measurement	Accuracy	Complexity		Capital cost per unit	Data post-processing
				installation	calibration		
WiM capacitive mat	pavement	capacitance of flexible membrane proportional to wheel forces	high	high	moderate	very high	moderate
WiM strain gauge	pavement	strain proportional to wheel forces	high	low	moderate	low	moderate
WiM piezo	pavement	voltage generated by piezo element proportional to wheel forces	moderate	moderate	moderate	low	moderate
strain gauge	bridge	x and $\dot{x}$. HV wheel forces and their dynamic derivatives	high	high	moderate	low	moderate
strain gauge	vehicle testing machine	x and $\dot{x}$. HV wheel forces and their dynamic derivatives	high	high	high	low	moderate

Instrumentation	Position	Measurement	Accuracy	Complexity		Capital cost per unit	Data post-processing
				installation	calibration		
LVDT	connected between bridge and ground	x and $\dot{x}$. HV wheel forces and their dynamic derivatives	very high	very high	moderate	high	low
String potentiometer	connected between bridge and ground	x and $\dot{x}$. HV wheel forces and their dynamic derivatives	moderate	moderate	moderate	low	low
Spring-and-wire	connected between bridge and ground	x and $\dot{x}$. HV wheel forces and their dynamic derivatives	high	moderate	moderate	moderate	low
Accelerometer	vehicle testing machine	$\ddot{x}$. HV wheel forces[4]	high	low	low	high	moderate
	bridge	x, $\dot{x}$ and $\ddot{x}$. HV wheel forces and their dynamic derivatives[4]	high	high	low	high	moderate

6 Evaluation of HV suspensions - derived measures

6.1 Temporal measures

From 1958 to 1960, the American Association of Highway Officials (AASHO) conducted testing on purpose-built road pavements. Approximately 1.1 x 10^6 axle repetitions occurred from US Army trucks being driven at 56km/h by drivers commissioned for the task. Arising from these tests, the “fourth power rule” was determined empirically, *viz*; pavement damage was proportional to the 4^{th} power of the static load of an axle (Cebon, 1999). This approach, whilst used almost universally for flexible pavement design has been criticised (Cebon, 1987, 1999) in that it does not take into account the concentration of dynamic loads at certain points over a length of road, effectively averaging the HV “bounce” forces into the empirical data (de Pont, 1992).

The following measures for HV suspensions are denoted ‘temporal’ (Cebon, 1987) since they are dependant on the forces on the chassis or wheels within a particular history.

6.1.1 Damping ratio

Apart from other specified parametric thresholds, HV road friendly suspensions are required to have:

- a damping ratio, zeta (ζ) of greater than 0.2 or 20% with dampers fitted; and
- a contribution of more than 50% to the overall viscous damping value of the measured damping ratio from the dampers

(Australia Department of Transport and Regional Services, 2004a).

The damping ratio (Section 2.5) in the context of a HV suspension is used to show how quickly a HV body returns to steady-state motion after encountering a bump in the road. Damping ratio is denoted by the Greek letter zeta (ζ), is dimensionless and usually shown as a number under 1 (e.g. 0.3) or a percentage (e.g. 30%) denoting the damping present in the system as a fraction of the critical damping value (Doebelin, 1980).

6.1.2 Damped natural frequency

For a HV suspension to be defined as “road-friendly”, not only does the damping ratio have to exceed certain values (above), but also the damped free vibration frequency (f) of body-bounce needs to be less than 2.0Hz.

Many researchers have derived the damped fundamental frequency of the body-bounce and axle-hop from frequency-domain analysis of the of transducer outputs mounted either on a HV or from wheel-force sensors in or on the pavement (Brannolte *et al.*, 2002; Gyenes & Simmons, 1994; Hahn, 1987a, 1987b; Middleton & Rhodes, 1994; Potter *et al.*, 1997).

The magnitude of body-bounce, wheel-force or other parameters of interest has been shown either as a frequency series using the output of a FFT (Davis & Sack, 2004, 2006; Sweatman, 1983) or as a power spectral density (PSD) against frequency (Gyenes & Simmons, 1994; Jacob & Dolcemascolo, 1998; LeBlanc & Woodrooffe, 1995; OECD, 1998). Essentially, the two methods (PSD *vs.* FFT) do not differ in their application to finding resonant frequencies. The vertical scale differs in that the vertical axis of the PSD graph is proportional to the square of the magnitude of the FFT graph (Vernotte, 1999).

6.1.3 Dynamic load coefficient

Sweatman (1983) developed a measure denoted the dynamic load coefficient (DLC) in his work “A study of dynamic wheel forces in axle group suspensions of heavy vehicles. Special Report No. 27” (Sweatman, 1983). This was, in part, based on earlier work (Sweatman, 1980) and was to account for, and allow comparison between, the relative effects of dynamic wheel-force behaviour of differing suspension types.

The dynamic load coefficient (DLC) was defined as the coefficient of variation of dynamic wheel forces relative to the static wheel-force; i.e. the coefficient of variation of the total wheel load.`

That approach utilised the concept that a measure of road damage could incorporate a damage component due to:

- dynamic forces present from wheel loads; plus
- a damage component due to the static forces present.

This was developed as the ratio of a measure of variation in dynamic wheel-forces to static wheel force. The static wheel-force was represented in this measure by the "mean wheel load" F_{mean} (Figure 7). The dynamic forces were represented in this measure as the standard deviation (σ) or root-mean-square (RMS) of the dynamic wheel-force (Figure 7).

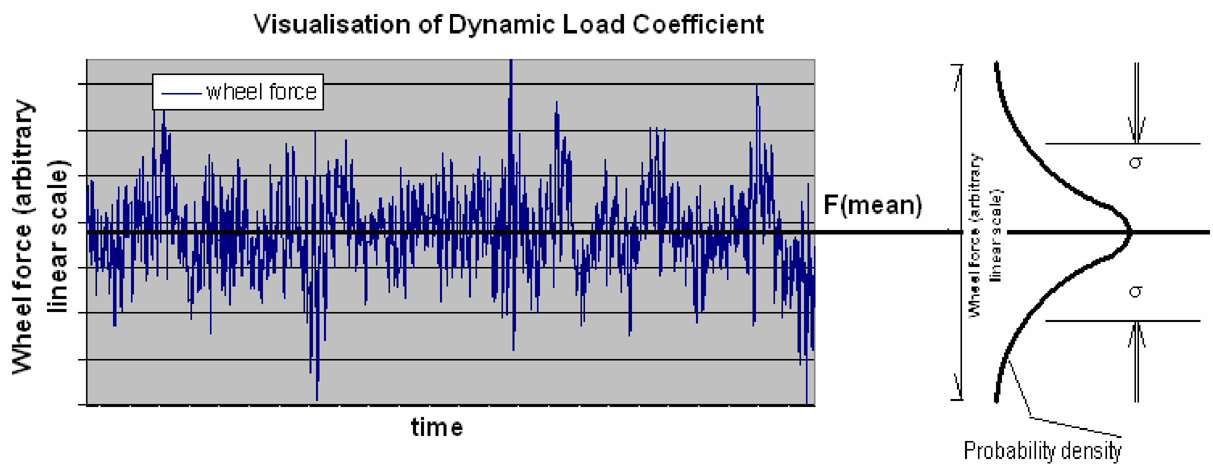


Figure 7. Summary of DIVINE report illustration for dynamic load coefficient (OECD, 1998).

The DLC may be defined mathematically, *viz*:

$$DLC = \sigma / F_{mean}$$

Equation 14

Where:

σ = the standard deviation of wheel-force; and

F_{mean} = the mean wheel-force (Sweatman, 1983).

It assumes that:

- dynamic loads are random;
- dynamic loads have a Gaussian distribution about F_{mean} as shown in Figure 7;
- road damage is distributed evenly along a length of road (Collop & Cebon, 2002); and
- road damage is proportional to the 4^{th} power of wheel-load.

Sweatman used various measures against which DLCs were plotted. These included averaging DLCs over all the runs made, regardless of the road surface (Sweatman, 1980, 1983) and against specific determinations of roughness, e.g. “smooth” and “rough” roads (Sweatman, 1983).

Differences in interpretation of the denominator in the DLC formula have been evident (de Pont, 1992). Both “static wheel-force” and “mean wheel-force during testing” have been used as the denominator (Potter *et al.*, 1997; Sweatman, 1983). It is for noting that Sweatman (1983) defined DLC with F_{mean} as the denominator. Other work (Potter *et al.*, 1997) redefined the DLC denominator to be the static force, (F_{stat}) on the wheel. There is a subtle but distinct difference between the two approaches. If the static tests are made on the wheel-force on level ground, the measured value will differ from on-road measurements since the camber of the road will place the CoG of the vehicle slightly to one side. F_{mean} will therefore differ from F_{stat}, depending on the angle of camber. It will also vary depending on the load-sharing ability of the suspension in question (de Pont, 1992).

Under DLC evaluation, a perfect suspension would have a DLC of zero. The range, in reality, is somewhere between 0 and 0.4 (Mitchell & Gyenes, 1989). Many researchers (Gyenes *et al*., 1992; Mitchell & Gyenes, 1989) have used DLC as one measure to differentiate suspension types from each another (e.g. steel *vs.* air). Despite this, the use of DLC has been criticised for purposes of attempting to distinguish between the damage potential of suspensions with different axle groups

(Potter, Cebon, Collop, & Cole, 1996) and despite being adopted as the de-facto standard as a road-damage determinant (OECD, 1998).

DLC continues to be criticized, most recently by Dr. Cebon at the 5th Brazilian Congress on Roads and Concessions; along the line: “how this [DLC] method leads to false conclusions regarding where and how to use road maintenance funds, spatial repeatability of road surface stress being the key issue.” (Lundström, 2007). This criticism is not new (Cebon, 1987).

6.1.4 Load-sharing coefficient

Early attempts to determine load-sharing of HV suspensions (Sweatman, 1976) were by measuring the load under a 40mm plank with a test HV driven over it to determine the changes in axle loads when compared with static loads.

Sweatman (1983) attempted to quantify the load-sharing ability of a multi-axle group in a number of ways, amongst which was the load-sharing coefficient (LSC). This was designed to be a measure of how a suspension group shared the total axle group load across the axles within the group. It is a value of the ability of a multi-axle group to distribute its load over each tyre &/or wheel in that group during travel.

The original definition of LSC was:

$$LSC = \frac{2 \times \mathrm{n} \times \mathrm{F}_{\mathrm{mean}}}{\mathrm{F}_{\mathrm{group\ (stat)}}}$$

Equation 15

Where:

n = number of axles in the group;

$F_{group\ (stat)}$ = axle group static force and

F_{mean} = the mean wheel-force in Figure 7 (Sweatman, 1983).

Note that this approach treated the load-sharing as being between axles.

Sweatman went on to state p6, (Sweatman, 1980) that the net increase in road damage [say, Δ_{damage}] due to unequal loading of (say) 10% *between axles* in a tandem group assuming, again, the '4th power law', may be calculated by:

$$\Delta_{damage} = 0.5 \times [1.1^4 + 0.9^4 - (1\text{-}1)] \times 100\%$$

Equation 16

This approach did not necessarily agree with other, early definitions such as that of Stevenson & Fry (1976) p24, that a HV with a "load equalising system" meant that an axle group utilised a suspension with the same spring types on each axle and that this resulted in "substantially equal sharing by all the ground contact surfaces of the total load carried by that axle group". Note the emphasis on wheel forces in the context of "ground contact surfaces", not axle forces.

LSC has been simplified and modified more recently to:

$$LSC = \frac{F_{mean}(i)}{F_{stat\ (nom)}}$$

Equation 17

Where:

$F_{stat\ (nom)}$ = Nominal static tyre force $= \frac{F_{group\ (total)}}{n}$;

$F_{group\ (total)}$ = Total axle group force;

F_{mean} (i) = the mean force on tyre/wheel i ; and

n = number of tyres in the group

(Potter, Cebon, Cole *et al.*, 1996).

Equation 15 and Equation 17 differ in that the latter focuses on the equalisation of wheel forces and the former on equalisation of axle forces (de Pont, 1992). This may be allocated to a difference in interpretation between schools of road damage: the vehicle modellers *vs.* the pavement modellers.

Potter *et al.*, (1996) examined variations in quantitative derivation of measures to describe the ability of an axle group to distribute the total axle group load. That work indicated a judgement that inter-axle relativities were the key to inter-wheel load-sharing.

The worth of the LSC as a prime determinant of suspension behaviour has declined but it is still used when describing the ability of a multi-axle group to distribute its load across all the wheels in its group.

6.1.5 Dynamic load-sharing coefficient

The original Sweatman research which examined different LSCs per suspension type instrumented only one hub per vehicle due to the cost (Sweatman, 1983). That work derived wheel-forces in multi-axle groups by taking the complement of measured wheel-load. Whilst understandable in terms of expense, inferring the other wheel loads as a complement of the measured load somewhat contradicted earlier work which stated, p5 (Sweatman, 1980): "…instantaneous axle forces will tend to be unequal due to dynamic forces generated over the road profile". If the wheel-forces were only out-of-phase, and there were no in-phase, common-mode or random wheel forces present, then that approach would have been valid.

Accordingly, the original research into LSC was questionable. de Pont (1997) also noted that dynamic load-sharing had not been addressed adequately and proposed a modification to the concept of load-sharing which took into account the dynamic nature of wheel-forces and any load-sharing which may occur during travel, denoted the dynamic load-sharing coefficient (DLSC):

$$\text{DLSC} = \sqrt{\frac{\sum (DLS_i - \overline{DLS_i})^2}{k}}$$

Equation 18

Where:

Dynamic load-sharing (DLS) at axle i, $DLS_i = \dfrac{nF_i}{\sum_{i=1}^{i=n} F_i}$

Equation 19

n = number of axles;

F_i = instantaneous wheel-force at axle i ;and

k = number of instantaneous values of DLS, i.e. number of terms in the series

(de Pont, 1997).

It is noted from Equation 18 that DLSC is the standard deviation of the dynamic load-sharing function, DLS*i*. Whilst this approach is an evolution from assumptions regarding complementary wheel-loadings and more inclusive of random, in-phase or common-mode relative excitation between consecutive axles, it does not consider that an axle can have differing wheel-loads at either end. This since the instantaneous wheel forces at axle i are summed to get F_i for comparison with the other axle/s in a multi-axle group. Again, there is an emphasis on inter-axle comparison. However, this approach could be applied to consecutive wheels in groups.

6.1.6 Load difference coefficient

de Pont also developed an alternative measure of load-sharing denoted the load difference coefficient (LDC) by examining the difference in variances of the wheel-loads between two axles. Its derivation is somewhat simpler that the DLSC:

$$\text{LDC} = \sqrt{\frac{Variance(F_i - F_{i+1})}{2F_{stat}}}$$

Equation 20

Where:

F_i = instantaneous wheel-force at axle i; and

F_{stat} is the static wheel-force.

It is noted that this was developed for a tandem axle group and treated loads per axle, not per wheel. However, with some work, it could be applied to consecutive wheels or axles in groups with three or more axles.

6.1.7 Peak dynamic wheel force

The peak dynamic wheel force (PDWF) is the maximum wheel-force experienced by a wheel during dynamic loading in response to a step (up or down) input (Fletcher *et al*., 2002). This measure is important as a link between analysis of wheel-force history and the work which promotes spatial repetition (Section 6.2) of HV wheel-forces as a measure of damage (Cebon, 1987; Collop & Cebon, 2002; Potter *et al.,* 1997; Potter *et al.,* 1995). In an alternative view that includes non-Gaussian wheel-force distributions in the spatial domain, PWDF may form part of a damage model applied to those points of maximum force on the pavement. When applied to historical data of wheel-forces on a particular length of pavement, the peak dynamic wheel-force may be used as an indicator of potential damage when raised to the appropriate power.

6.1.8 Peak dynamic load ratio (dynamic impact factor)

One of the criticisms of DLC is that it assumes that a Gaussian distribution of wheel forces in the time domain will be Gaussian as a spatial variable. Where the wheel-forces may not be Gaussian (which suggests an alternative to DLC) and when considering longitudinal position variable-space, PDLR may be considered. It is the ratio of the maximum wheel-force experienced by a wheel during dynamic loading to the static wheel-force:

$$\text{PDLR} = \frac{PDWF}{F_{stat}}$$

Equation 21

Where:

PDWF = peak dynamic wheel-force measured instantaneously during the test (Section 6.1.7); and

F_{stat} is the static wheel-force.

It is not based on a particular distribution and is useful when comparing data with similar distribution sets (Fletcher *et al*., 2002).

A similar measure for axle forces has also been denoted “dynamic impact factor” (DIF) and used in the earlier evaluations of different types for suspensions for road damage:

$$\mathrm{DIF} = \frac{PDF}{F_{stat(axle)}}$$

Equation 22

Where:

PDF = peak instantaneous force measured during the test; and

$F_{stat\ (axle)}$ is the static axle force

(Woodrooffe & LeBlanc, 1987).

Again, we see that there is a difference in philosophy between the allocation of road network asset damage to axle forces or wheel-forces.

6.1.9 Road stress factor

In attempting to account for the effects of dynamic loadings above static wheel forces, Eisenmann (1975) developed the concept of a “road stress factor” Φ:

$$\Phi = \mathrm{K}\mathrm{F}_{\mathrm{stat}}^{4}\ [1 + 6\,(\bar{\mathrm{s}})^{2} + 3(\bar{\mathrm{s}})^{4}]$$

Equation 23

Where:

F_{stat} = mean axle load;

$\bar{s}$ = the coefficient of variation in dynamic wheel load; and

K = a constant.

A dynamic road stress factor ν which accounts for damage to roads due to dynamic effects was defined and developed thus (Sweatman, 1983):

$$\nu = \frac{\Phi}{KF_{stat}^4}$$

Equation 24

=> $\nu = 1 + 6(\bar{s})^2 + 3(\bar{s})^4$

Equation 25

In developing this measure, the following assumptions were made:

- dynamic loads are random;
- dynamic loads have a Gaussian distribution about a mean equal to F_{mean} as shown in Figure 7; and
- road damage is proportional to the 4^{th} power of wheel-load (de Pont, 1992).

Sweatman (1983) used the "fourth power rule" and the estimation of road damage quantified by the "road stress factor" from Eisenmann's equation (1975) to further develop the concept of "dynamic road stress factor", denoted simply as road stress factor or RSF. It was given by:

$RSF = (1+6DLC^2 + 3DLC^4)\ F^4_{stat}$

Equation 26

Where:

DLC is as defined in Equation 14; and

F_{stat} = the static wheel-force.

Note that this formula equates $\bar{s}$ = DLC which is not strictly correct given the DLC was derived from the standard deviation of the wheel forces *per test* divided by the mean wheel-force for *all tests* (Sweatman, 1983).

In addition to this assumption, Equation 23 assumes that the road damage is not only Gaussian in time but also Gaussian in space; i.e. distributed evenly along the length of road (Collop & Cebon, 2002).

All of these variations were attempts to account for the road damage from dynamic wheel forces due to the 4th power of the instantaneous (i.e. dynamic) component of HV wheel forces. This formula and others of similar form have been denoted dynamic pavement wear factor (DPWF) or dynamic pavement stress factor (DPSF) (Gyenes *et al.*, 1994; Gyenes & Simmons, 1994; Mitchell & Gyenes, 1989). Sometimes the formula is normalised by either removing the F^4_{stat} factor or by dividing by it when providing relative RSF measures between one suspension and another.

6.1.10 95%ile road stress factor

Sweatman (1983) went on to develop a measure for large (95%ile) but infrequent forces from dynamic wheel loads which included a spatial repetition component. Again he used the "fourth power rule" (Eisenmann, 1975) as a basis for a measure of road stress. The formula used was:

$$RSF_{95} = (1 + 1.645DLC)^4$$

Equation 27

where DLC is as defined above (Equation 14).

Both the RSF and RSF_{95} have been criticised (Cebon, 1987, 1999; Cole *et al.*, 1996; Lundström, 2007) because they do not account for spatial repeatability. Spatial repeatability is the predisposition of HVs with similar suspension characteristics to cause impacts to the pavement at the same point after encountering a bump. The RSF and the RSF_{95} also rely on the validity of the 4^{th} power law. Harmonisation of the two approaches was proposed by de Pont *et al.*, (de Pont & Pidwerbesky, 1994).

6.1.11 Dynamic increment

The dynamic increment is the incremental value or fraction above the static load that is imparted to bridges by dynamic forces of vehicles.

Various researchers such as Heywood defined the DI as:

$$DI = \frac{(\delta_{dyn} - \delta_{static}) \times 100\%}{\delta_{static}}$$

Equation 28

Where:

δ_{dyn} = peak dynamic deflection or strain in the structure; and

δ_{static} = peak static deflection or strain in the structure

(Cantieni, 1992; Heywood, 1995).

DI is not a good indicator of bridge susceptibility to dynamic loadings since the diversity of bridges and their construction makes determination of dynamic behaviour above 7Hz difficult other than by empirical means (Heywood & Boully, 2000). Nonetheless, bridges with resonant frequencies greater than 9Hz and spans greater than 40m have DIs 5 times those of shorter-span bridges with low resonant frequencies (Heywood & Boully, 2000).

6.1.12 Dynamic load allowance

The dynamic load allowance (DLA) nominates an increment in the static design forces to allow for dynamic and resonant loads in bridge design. It is similar to, but more complex in its derivation than, dynamic increment. It is influenced by the dynamic interactions between vehicles and structures (Ashebo *et al.*, 2007) and is contained in the Austroads Bridge Design Code (ABDC). The 1992 ABDC did not show design DLA values for frequencies above 7Hz, excluding the potential for bridges to be assessed for axle-hop induced dynamic coupling between HVs and bridges. This approach has been criticised (Senthilvasan *et al.*, 2002). The 2004 update, AS 5100.2 – 2004, addressed this issue broadly in Table 6.7.2 by specifying a flat-rate DLA for all frequencies.

Australian research found that poor profiles leading up to and on a significant number of short-span bridges had the potential to cause axle-hop and vehicle/bridge dynamic coupling forces which exceed the (then) load allowances (Heywood & Boully, 2000; Sweatman & Addis, 1998). Coupling at axle-hop frequencies was weak to non-existent for bridges with natural frequencies of less than 8Hz (Moldoveanu & Heywood, 1997).

6.2 Spatial measures

6.2.1 General

Cebon (Cebon, 1987, 1993, 1999) has championed the concept of spatial damage assessment for dynamic wheel loads. This is an approach where the damage due to HV dynamics is quantified over a particular length of pavement. It is based on the probabilistic nature of road damage. It contains the concept that road damage leading to loss of serviceability occurs at only a small proportion of the length of road (Cebon, 1987). "Spatial" measures include weighted stress, aggregate force, strain fatigue damage and pavement deformation (Cebon, 1987; Collop & Cebon, 2002; Potter *et al.,* 1997; Potter *et al.,* 1995). These approaches break the pavement into a number of short segments to determine peak forces on that segment. It predicts the pavement damage due to HV wheel forces, sometimes using the highest frequency of interest in the process. These models require an intimate knowledge of pavement behaviour resulting from wheel forces.

Some studies have used instrumented pavements to correlate spatial and temporal measurements of wheel-loads (Cole *et al*., 1996; Gyenes & Mitchell, 1992). These, as well as other studies (Hahn, 1987a, 1987b; LeBlanc & Woodrooffe, 1995), have shown strong evidence that, for a specific type of vehicle at a specific speed, wheel-loads along a length of road are not distributed randomly but are concentrated at specific points on the length of road. This effect is known as "spatial repeatability". For specific classes of tested vehicles the spatial correlation of wheel-loads was reported as moderate-to-high and judged highly dependant on travel speed, HV suspension and chassis configurations (Cole *et al*., 1996; LeBlanc & Woodrooffe, 1995). The issue of allocating road damage at specific points on the pavement to the entire HV fleet became less clear, however, when attempts were made to correlate wheel loads measured from test HVs against spatial wheel-loads measured from the pavement. Diversity of suspension types, such as steel walking-beam or air-sprung, and diversity of speed reduces the correlation to moderate-to-low (LeBlanc & Woodrooffe, 1995). These lower correlations were noted even on a test semi-trailer tanker that was configured specifically to have its prime-mover and trailer suspensions replaced with steel or air for testing (LeBlanc & Woodrooffe, 1995). Increasing homogeneity of the parameters of the RFS HV fleet will nonetheless result in more highly-correlated wheel-forces. Spatial repetition will therefore need to be addressed (LeBlanc, 1995); suspensions with common parameters will bounce their wheels onto the same places on the pavement after encountering a bump.

When considering the introduction of HML into Australia, spatial repeatability measures contributing to pavement damage were acknowledged as an approach to determining road damage but not included in the methodology (National Road Transport Commission, 1993a).

Attempts have been made to harmonise the spatial damage models with temporal models that rely on measurements from HVs (de Pont & Pidwerbesky, 1994). This approach has not been adopted widely.

6.2.2 Aggregate force

Cebon (1987) developed the concept of aggregate force because of perceived difficulties with DLC. Specifically, DLC does not account for:

- differences in wheel loads between suspension types with different numbers of axles;
- the influence of dynamic wheel loads generated by other axles on the vehicle and transferred to the specific wheel in question; and
- the effect of lightly loaded axles with large DLCs *vs.* heavily loaded axles with small DLCs (Cebon, 1987).

These issues led to the development of an aggregate force (AF) measure that allowed for the spatial repeatability of wheel-forces to be accommodated. It requires the use of an instrumented pavement of *n* segments with a sensor in each segment.

The aggregate force F at sensor k may be derived using instantaneous wheel forces on that segment of the pavement:

$$F_k = \sum_{j=1}^{N_a} P_{jk} \qquad \text{for k = 1, 2, 3, … } n$$

Equation 29

Where:

F_k = instantaneous wheel-force at sensor k;

N_a = number of axles on the vehicle; and

P_{jk} = the force applied by wheel or tyre j to sensor k

(Cebon, 1987; Cole *et al*., 1996).

6.2.3 Weighted aggregate force

To find a measure of damage to a pavement using any of the pavement damage "power rules" the aggregate force F_k may be raised to the appropriate power q. For a length of instrumented pavement, the aggregate force F at sensor k may be raised to the appropriate power q to determine road damage due to instantaneous wheel forces on that pavement:

$$F_k{}^q = \sum_{j=1}^{N_a} P^q{}_{jk} \qquad \text{for k = 1, 2, 3, … } n$$

Equation 30

Where:

F_k = instantaneous wheel-force at sensor k;

N_a = number of axles on the vehicle;

P_{jk} = the force applied by wheel or tyre j to sensor k;

q = the selected exponent of road damage.

This is termed the weighted aggregate force (Cebon, 1987).

For permanent deformation, such as rutting, $q = 1$ may be appropriate (Gillespie *et al*., 1993) or the "fourth power rule" where $q = 4$ for flexible pavements may be used (Cole *et al*., 1996). Similarly, any other damage exponent (such as 12 for concrete pavements) can then be chosen. When $q = 4$, this measure may be comparable with Sweatman's (1983) "road stress factor" (Cebon, 1987).

6.2.4 Correlation coefficient

For any two dynamic wheel load measurements, their correlation function can be used determine the measure of spatial repeatability. For highly-correlated wheel-force histories measured along tyre-paths during testing, the maxima and minima will tend to occur at similar points on the travelled path.

This effect can be defined using a measure defined as the correlation coefficient (ρ):

$$\rho \propto \frac{[f_n(t) - \overline{f_n(t)}][f_{n+1}(t) - \overline{f_{n+1}(t)}]}{\sigma_{fn}\sigma_{fn+1}}$$

Equation 31

Where:

ρ = the correlation coefficient;

f_n (t) = the n^{th} wheel-force time history signal;

$\overline{f_n(t)}$ = the mean of the n^{th} wheel-force time history signal; and

σ_{fn} = the standard deviation of the n^{th} wheel-force time history signal.

Correlation coefficients greater than $\sqrt{2}/2$ indicate that the wheel-forces are highly correlated and that spatial repeatability is occurring. This leads to the conclusion that the pavement will fail at specific patches for the suspension under test at the test speed (Kenis *et al*., 1998).

6.2.5 Spatial repeatability index

Combining the concepts of aggregate force with correlated forces, Collop *et al.*, (1994) developed the spatial repeatability index (SRI). This was defined as the correlation coefficient of the aggregate tyre force histories of two signals. Mathematically, it may be defied as:

$$SRI = \frac{\dfrac{\omega}{2\pi V}\displaystyle\int_0^{\frac{2\pi V}{\omega}} [f(t) - \overline{f(t)}][g(t) - \overline{g(t)}]dx}{\sigma_f \sigma_g}$$

Equation 32

Where

SRI = spatial repeatability index;

t = x/V;

Equation 33

x = the distance along the measured wheel path/s;

V = the velocity of the vehicle;

f(t) = h(t) + ε_f(t) representing a reference wheel-force time history signal with frequency ω and an error signal ε_f ;

g(t) = k(t) + ε_g(t) representing a test wheel-force time history signal with frequency ω and an error signal ε_g ;

h(t) = sin(ωt), a reference wheel-force time history signal as an assumed sinusoid;

k(t) = sin(ωt + θ), a test wheel-force time history signal as an assumed sinusoid with phase-shift θ;

σ_f = the standard deviation of the wheel-force time history signal f;

σ_g = the standard deviation of the wheel-force time history signal g;

$\overline{f(t)}$ = the mean of f(t); and

$\overline{g(t)}$ = the mean of g(t)

(Collop *et al*., 1994).

6.2.6 Spatial distribution number

The spatial distribution number (SDN) was defined as the standard deviation of the SRI. The SDN was derived to describe the degree of spatial repeatability present in the entire HV fleet (Collop, Cebon, & Cole, 1996):

$$\text{SDN} = \sqrt{\frac{\sum(SRI - \overline{SRI})^2}{n}}$$

Equation 34

Where:

SDN = spatial distribution number;

SRI = spatial repeatability index; and

n = number of SRI measures.

Perfect repeatability would yield a SDN of zero and totally random wheel-force distributions would give a SDN of ∞ (Collop *et al*., 1996).

6.3 Summary of this section

The following table is a broad summary of the measures described in this section for defining dynamic behaviour of HVs. This is not necessarily definitive since the use of these measures is still the subject of robust discussion amongst researchers (Lundström, 2007). They have been included in this section and summarised here as a prelude to the Findings section which explores the differences and indeterminacy of the measures available. Ultimately, all measures are used to estimate HV damage to the road network asset.

Table 3. Summary of the measures for defining dynamic behaviour of HVs

Measure	Acronym/ symbol	Use	References
Damping ratio	zeta, ζ	In the context of a HV suspension, defines the level of control over subsequent body-bounce and axle-hop excursions after a truck hits a bump. Early work in Germany led to the adoption of a lower bound for damping ratio values for “road-friendly” HV suspensions. Higher damping ratios bring the excursion after a bump in control faster than for lower ζ.	(Australia Department of Transport and Regional Services, 2004a; Chesmond, 1982; Doebelin, 1980; Eykhoff, 1974; Meriam & Kraige, 1993; Thomson & Dahleh, 1998)
Body-bounce	Hertz, Hz	Defines the frequency of the body-axle interaction for a HV. Early work in Germany led to the adoption of low body-bounce frequency values for “road-friendly” HV suspensions.	(Australia Department of Transport and Regional Services, 2004a; Chesmond, 1982; Doebelin, 1980; Eykhoff, 1974; Meriam & Kraige, 1993; Thomson & Dahleh, 1998)

Dynamic load coefficient	DLC	A measure of variation (i.e. std. dev.) in dynamic wheel-forces when compared with static wheel force. This to account for, and allow comparison between, the relative damage effects of dynamic wheel-force behaviour of differing suspension types.	(Eisenmann, 1975; Sweatman, 1980, 1983)
Load-sharing coefficient	LSC	A measure of the load-sharing ability of a multi-axle group. It is a value of the ability of a multi-axle group to distribute its load over each axle (or wheel) in that group during travel. Differing interpretations have been used based on axle *vs*. wheel loads and mean dynamic wheel force *vs*. static wheel force. LSC assumes that the load is always shared instantaneously between axles/wheels. Better load-sharing leads to less asset damage.	(de Pont, 1997; Potter, Cebon, Cole *et al*., 1996; Sweatman, 1983)
Dynamic load-sharing coefficient	DLSC	This measure was developed to address the dynamic nature of wheel-forces and load-sharing between axles during travel. Unlike LSC, this measure does not assume that the load is always shared instantaneously.	(de Pont, 1997)
Load difference coefficient	LDC	A similar alternative to DLSC using variance in loads between two axles. This method treats loads per axle.	(de Pont, 1997)
Peak dynamic wheel force	PDWF	The maximum wheel-force experienced by any HV wheel during a dynamic event. Can be used to determine instantaneous damage using the “fourth power rule”.	(Fletcher *et al*., 2002)

Peak dynamic load ratio	PDLR	Ratio of the maximum wheel-force experienced during dynamic loading of a wheel to the static wheel-force. Useful for comparing one suspension type with another to determine relative damage potential.	(Fletcher *et al*., 2002)
Dynamic impact factor	DIF	Similar to PDLR but this measure uses axle forces instead of wheel forces to differentiate between different types for suspensions for road damage.	(Woodrooffe & LeBlanc, 1987)
Road stress factor	RSF	This was the start of efforts to account for damage to roads due to dynamic effects. RSF contains a factor for damage due to dynamic loading by HV wheels and a factor for damage due to steady-state (static) loading by HV wheels. Used as the basis of other measures such as DLC. Relies on the validity of the "4^{th} power law" for unbound granular pavements, for instance.	(Eisenmann, 1975; Sweatman, 1983)
95%ile road stress factor	95% DLC	Developed as a measure to account for large (95%ile) but infrequent forces from dynamic wheel loads. Based on the "fourth power rule" of these infrequent events and the RSF philosophy, it allows "road stress" to be determined as a measure of road damage from large but infrequent dynamic loading. Relies on the validity of the "4^{th} power law".	(Eisenmann, 1975; Sweatman, 1983)
Dynamic increment	DI	Incremental value (or fraction) above the static load that is imparted to bridges by dynamic forces of vehicles. Used by bridge designers to understand the allowance required for dynamic loadings due to HVs.	(Cantieni, 1992; Heywood, 1995; Heywood & Boully, 2000)

Dynamic load allowance	DLA	An increment in the static design force for bridge design. It similar to, but more complex in its derivation than, DI in that it allows designers to compensate for dynamic interactions between vehicles and structures.	Austroads Bridge Design Code AS 5100.2 – 2004 and others (Ashebo *et al*., 2007; Heywood & Boully, 2000; Moldoveanu & Heywood, 1997)
Aggregate force	AF	Road damage is probabilistic under this measure. AF then goes to the concept that road damage leading to loss of serviceability occurs at only a small proportion of the length of road. AF attempts to quantify this damage by examining the point at which the largest wheel forces occur on a particular length of pavement.	(Cebon, 1987, 1993, 1999; Cole & Cebon, 1989, 1992; Cole *et al*., 1992, 1996; Collop *et al*., 1996; Collop *et al*., 1994; Potter, Cebon, Cole *et al*., 1996; Potter *et al*., 1994)
Weighted aggregate force	WAF	Based on AF, this pavement damage measure raises AF to a chosen road damage exponent. This may be (say) 4, if the "fourth power rule" is chosen to determine damage due to HV wheel forces at a particular point on a stretch of road.	(Cebon, 1987; Cole *et al*., 1996)
Correlation coefficient	CC	This measure allows two wheel-force histories to be compared to determine if they are correlated. If they are similar, damage is more likely to occur at localised portions of pavement.	(Kenis *et al*., 1998)

Spatial repeatability index	SRI	SRI combines CC and AF to allow comparison between a test suspension and a reference suspension. This to determine a measure of spatial repeatability between a test HV and a reference HV.	(Cole & Cebon, 1992; Collop *et al*., 1996; Collop *et al*., 1994; Kenis *et al*., 1998)
Spatial distribution number	SDN	This measure is the std. dev. of the SRI. It provides an indication of how spatially correlated is a fleet of HVs. This to determine a measure of spatial repeatability (and the probability of localised damage) from a given fleet of HVs.	(Collop *et al*., 1996)

7 Findings

This report was generated in preparation for the NTC in-service testing of HV suspensions project. This section contains the findings.

7.1 Vehicles *vs.* pavements – where to measure forces?

There were found, broadly, two approaches to determination of HV wheel forces:

- Vehicle models; and
- Pavement models.

The vehicle model school tended to think of vehicle dynamics in terms of inter-axle force relationships. This was evident in the early work that treated dynamic load-sharing as a phenomenon involving sequential axles, for example.

Pavement models (and modellers) did not particularly concern themselves with this approach – they incorporated wheel-forces by examining the damage done to pavements directly: where the tyre meets the road.

7.2 Pavement models

The pavement models were further broken down into differing approaches for road and pavement damage:

- the Gaussian - a random distribution of HV wheel forces on a road length with all pavement segments subject to a roughly evenly-distributed probability of road damage; or
- the Spatial - the characteristics of HV suspensions and travel speeds incline wheel forces to occur at specific locations on any length of road and that road damage is therefore concentrated at those specific points.

7.2.1 Gaussian distribution

This approach assumes that HV wheel-forces are randomly distributed along a stretch of road. It relies on measurement of wheel-force histories (statistical analysis) and development of road-damage models based on (say) the "fourth power rule".

As noted in Section 6.2.1 this was the approach that the NRTC took when considering the introduction of HML into Australia (National Road Transport Commission, 1993a).

Some allowances have been made for dealing with quasi-spatial repetition issues such as developing 95th%ile formulae for large magnitude but low-probability wheel-force events (Sweatman, 1983).

7.2.2 Spatial repeatability

The second philosophical school of road-damage focuses on the road asset, measures wheel forces directly and develops road-damage models that account for dynamic loads as HVs travel over segments of pavement. The approach of capacitive mats and, indeed, all in-road sensor-based technology proceeded on the basis that only by measuring the actual wheel-forces at the road surface could the damaging effects of wheel-forces be determined accurately.

The central assumption, backed by not inconsiderable empirical data, is that the pavement will fail due to localised peak dynamic forces damaging the pavement at certain repeated points along a length of road. Because of the concentration of wheel forces creating localised pavement failure, it proposes that network utility is reduced by denying service to an entire length of road (Cebon, 1987, 1993, 1999). It rejects the use of averaged dynamic forces as a predictor of pavement damage (Potter *et al*., 1994).

7.2.3 Spatial repeatability *vs.* Gaussian distribution

Many researchers have promoted one, the other or parts of both of these approaches.

There are some disadvantages with spatial measurement. One is the need for long lengths of instrumented road, up to 250m (Potter *et al*., 1997). Another is the inability to measure conveniently a variety of roads with varying surfaces. Instrumented vehicles can record data from on-road excitations for longer distances than an instrumented pavement. On-vehicle measurement ability is more portable, allowing recording of different road surfaces more conveniently. Further, since road profiles alter over time, the wheel-forces, and therefore road-damage, can only be determined using instrumented pavement methods for the particular set of circumstances at the time of the testing. Repeatability would require the same length of pavement to be measured periodically. This is easier with an instrumented vehicle than an instrumented pavement.

Spatial measures attempt to deal with rutting damage in a different manner from fatigue damage. Rutting is the result of repeated passes of wheels and is related to speed and vehicle static mass. Fatigue is a result of many individual dynamic wheel forces impacting the pavement at the same or similar points along a length of road (Potter *et al.*, 1995).

Whilst there may be medium-strong spatial correlation of wheel-forces for particular vehicles at particular speeds (Cole *et al*., 1996), that correlation reduces to moderate-to-low for the fleet. This due to different suspension types and different speeds of operation across the fleet. Accordingly, Eisenmann's formula underestimates pavement wear (Gyenes *et al*., 1994) and the 95th%ile formula overestimates pavement wear from dynamic wheel loads (LeBlanc & Woodrooffe, 1995). That the dominant vehicle types have *approximately* Gaussian axle load distributions and therefore may have their wheel forces characterised by mean and standard deviations has, however, been validated in more recent work (de Pont, 2004). This assurance was based on DIVINE project WiM data and incorporated the empty state as well as the laden state for vehicles representing a greater majority of HVs on Australian roads.

If spatial measures are valid in predicting pavement failures then peak dynamic forces will cause these failures (Cole *et al*., 1996; Collop *et al*., 1994). Even if they are not spatially correlated, the "power law" model for wheel-force damage will still indicate pavement failure. The difference between these two models appears to be the number of repetitions before failure rather than whether to choose average dynamic forces over peak ones.

As pointed out by critics of methods that assume Gaussian distribution of wheel-forces, the "fourth power rule" was developed with dynamic loadings already "built into" the AASHO experimental data. To exploit spatial repetition as a pure approach, both steady-state wheel-force load and dynamic wheel-force load should be separated and pavement damage treated as two separate phenomena: rutting and fatigue respectively. In a perfect world, the rutting and the fatigue damage predictors would be combined *after* this point in the process to form a composite damage model. As pointed out by the critics of Gaussian wheel-force models, this is in contradistinction to the "one-size-fits-all" lumped empirical formula containing both concepts. Nonetheless, and with some awkward logic, spatial models of pavement damage still adopt the 4th power exponent for their damage predictions (Collop *et al*., 1994) of flexible pavements.

If dynamic and the steady-state forces were separated from the original AASHO data, it may well be that the failure modes and associated exponents would be:

- 1 - 2 (probably 1.4) for rutting of NZ roads;
- 2 for fatigue and between 3.3 – 6 for rutting on Australian roads;
- 3.3 for fatigue and 4 for rutting on Finnish roads;
- 2 for fatigue and 8 for rutting on French roads;
- 1.2- 3 for fatigue on Italian roads;
- 1.3 – 1.9 for fatigue and 4.3 for rutting on USA roads

depending on tyre configurations and axle combinations used in the country (de Pont & Steven, 1999; Pidwerbesky, 1989). This as opposed to the dynamic forces that produce point-failure of flexible pavements which gave the empirical "4th power rule" historically.

In summary, whilst the 4th power rule has been questioned, it remains the basis of measures to determine flexible pavement damage.

7.3 Vehicle models

7.3.1 Defining road damage from a vehicle-based framework

Mitchell & Gyenes (1989) noted that one measure alone was not sufficient to determine the road-damaging potential of a particular suspension and concluded that a judgement based on:

- LSC;
- DLC;
- low and high frequency vibration forces (body-bounce and wheel hop); and
- peak axle loads

would be necessary since the measures developed up to that point described different suspension parameters, depending on behaviour.

Early work to define characteristics of HV suspensions that might be less damaging to the road network asset proposed measures other than frequency and damping ratio. These included load-sharing, cessation of the use of wide tyres &/or dispensing with centrally-pivoted suspensions (e.g. RHS, Figure 2) with inadequate damping values by design (Cole & Cebon, 1991). The first and last of these were proposed by Sweatman (1983).

de Pont's work (de Pont, 1997, 1999) validated the concept of testing suspensions rather than whole vehicles by pointing out that a deterministic whole-of-vehicle test regime for road friendliness is neither practical nor meaningful given:

- the position of in-service loads on any given vehicle varies from trip to trip;
- any given prime-mover could be towing a range of trailers with either RFS at varying functionalities or no RFS fitted;
- axle loads vary (as, indeed, does the entire vehicle mass) depending on the utilisation of the vehicle from empty to full load; and

- the road friendliness (and dynamic pavement loading) varies with speed.

Cebon (1999) noted that changing one sub-system on an articulated vehicle (e.g. trailer or prime-mover) to a non-RFS unit can decrease the road-friendliness of the remaining part of the particular vehicle.

It may be valid to question, however, what should be the desirable parameters for RFS as measured by testing methods other than the EU way. Again, de Pont's work (1999) showed that the values measured for resonant frequencies, etc. at different loads and speeds do not vary significantly from those derived from the EU testing if the centre-of-gravity is placed over the particular suspension (component) under test. Other research pointed out that when determining suspension characteristics measured from a shaker-bed or reaction frame it was necessary to either test a single axle with the body fixed or test the whole vehicle. Testing selected axles did not reflect suspension nor vehicle performance (Stanzel & Preston-Thomas, 2000).

This infers that whole-of-vehicle testing is the only valid method but this:

- ignores the reality that prime-movers and trailers are rarely used continuously as a unit;
- is hard to reconcile against the supposed validity of individual axle tests; and
- does not seem to encompass de Pont's work (de Pont, 1999).

There have been three options for road authorities and regulators to assess "road-friendly" suspensions:

- design prescriptive for generic type (blanket design-approval);
- parametric definition; or
- performance based.

Design prescriptive (blanket design-approval)

Design approval allowed concessional mass increases for air suspensions in the UK without consideration of how these suspensions behaved dynamically (de Pont, 1997). The design-prescriptive approach is flawed and allowed poorly designed air-suspended vehicles onto UK roads (Cambridge University Transportation Research Group, 2005). It did not allow for good engineering. Even so, until the work of Cole & Cebon (1996; 2007) this approach had no basis to show that suspensions with high damping ratios and low spring rates, relative to conventional steel suspensions, were any less damaging. Further, that work was based on the use of the dynamic load coefficient (DLC) that has been open to discussion as an indicator of damage. This debate due to the issue that spatial repetition (see Section 6.2) not being well addressed in the DLC model.

Performance based

Performance based testing involves installing instruments on either the vehicle under test or the road surface and measuring the dynamic response to realistic operating conditions. This latter has been proposed as travel over a standardised road profile (Gyenes *et al.*, 1992) or by the use of a road simulator (de Pont, 1997). The performance based testing approach has not been implemented due to the cost and complexity of instrumenting a roadway, the cost of road simulators or the expense of installing instrumentation on individual HVs for the purposes of routine testing (de Pont, 1997).

Parametric testing

To counter the drawbacks of the other two approaches, parametric tests have been developed and these are now incorporated into the current Australian test regime for “road-friendliness”. These tests have shock absorber characteristics integral to the outcomes of the tests and allow different directions for test perturbations. Sweatman *et al.*, (2000) and Prem *et al.*, (1998) documented the force/displacement relationships for shock absorbers depending on direction of travel. Prem *et al.*, (1998) noted variations in the derived values for damping ratio, dependent on the excitation method due to, in part, this non-linearity. That this phenomenon created a 60% variation in derived damping ratio when comparing a "lift and drop" test with the EC step down test has been documented (Uffelmann & Walter, 1994). With regard to the EC type-testing regime, it is therefore questionable to have the results of a "pull up" test compared with, and equated to, a "pull down" test. Accordingly, Prem *et al.*, (1998) discounted pull up as equivalent to pull down testing for in-service testing of suspensions.

7.4 "Road-friendliness" of HV suspensions

Well-designed air-spring suspensions have low Coulomb friction, low spring rates and viscous damping which exhibits required hysteresis (Sweatman & Addis, 1998). Conventional steel suspensions are characterised by:

- high spring rates;
- hysteresis dependant on inter-leaf Coulomb friction which "breaks" when a large-enough bump or a rough-enough road is encountered (O'Connor *et al*., 1980; Whittemore, 1969); and
- damping which is less dependant on the shock absorbers and more on the unpredictable inter-leaf Coulomb friction, which is a function of both the size of the bump and the roughness of the road exercising the springs (OECD, 1992).

Air suspensions fitted to prime-mover drive axles:

- increased wheel forces for all trailer suspension types when compared with steel-suspended prime-movers and
- often caused higher whole-of-vehicle road damage when compared with steel-suspended prime-movers (Potter *et al.*, 1995).

The Cambridge web site (Cambridge University Transportation Research Group, 2005) states, with regard to the large-scale capacitive mat testing:

> "One interesting discovery was that some of the highest dynamic tyre forces were generated by vehicles with (so called) 'road-friendly' air suspensions. The reason for this apparently anomalous result is that air suspensions require well maintained shock absorbers to provide their damping - otherwise their bouncing motion can be lightly damped and they can generate very high dynamic loads.
>
> Existing EC regulations for road-friendly suspensions encourage the use of air suspensions by providing a 1 tonne payload incentive to operators. However, the regulations do not control their in-service dynamic loading performance, or suspension and damper maintenance. It seems, therefore, that current regulations have the opposite effect to that intended by the European Commission... In a significant proportion of cases, they increase rather than decrease dynamic loads and the resulting road damage."

The capacitive mat tests did not necessarily differentiate between compliant[9] air suspensions and those which did not meet the EC directive for reasons of poor design or lack of maintenance (Potter *et al.,* 1995).

It is therefore possible that the assumption that no steel HV suspensions were "road-friendly" and all air suspensions were "road-friendly" was flawed. Certainly the testing by Potter *et al.,* (Potter, Cebon, Collop *et al*., 1996) of different suspension types did not always produce consistently lower DLC and aggregate force results for air when compared with steel, particularly at higher speeds. Other research (Mitchell & Gyenes, 1989) did. Neither did air springs always come out any more favourably when compared with conventional steel in early testing for differences in road network asset damage done by differing types of suspensions (de Pont, 1997; Hahn, 1987a, 1987b; Magnusson, 1987; Potter *et al*., 1995; Woodroofe & Le Blanc, 1988). Other researchers such as de Pont recommended against such arbitrary design-based judgements, finding that not all air suspension were "road-friendly' (de Pont, 1997).

Certainly the diversity of research in this area has shown that, whilst use of conventional steel HV suspensions can result in higher dynamic loads (Simmons & Wood, 1990) and greater damage exponents (Cole & Cebon, 1995; 1996; 2007) than

[9] Fitted with shock absorbers meeting EC directive 85/3/EEC (as amended by Council directive 92/7/EEC).

air suspensions, this was not a consistent result for this type of HV spring media. Network asset damage is heavily dependant on suspension design and the ability of the suspension in question to equalise load (de Pont, 1997; Sweatman, 1980).

Whilst "road-friendly" air suspensions with lower spring rates and higher damping ratios than conventional steel suspensions may have lower dynamic forces in any given wheel-force history, air suspensions may not be any less damaging than conventional steel suspensions when the issue of spatial repeatability is taken into account (Potter, Cebon, Collop *et al*., 1996). This issue becomes particularly important for a highly-homogeneous HV fleet with similar fundamental frequencies (e.g. VSB 11-compliant) yielding highly correlated wheel-loads derived from almost-identical suspension wavelengths (Kenis *et al*., 1998). Further, within the "friendlier" air suspension class, when spatial measures such as 95th%ile aggregate force were examined, tandem axle groups generated more damage than tri-axle groups (Potter *et al*., 1994).

Pestrerev *et al.,* (2004) showed that when a truck's suspension is softened, for purposes of meeting the RFS criteria, the magnitude of body bounce forces is decreased. This increases the amplitude of axle-hop forces because softer springs do not control the axle as well as stiff springs. Hence, although air-suspended vehicles are considered to be "road-friendly", their axle hop characteristics can produce greater pavement damage when poorly damped. Along these lines Sweatman & Addis, p9 (1998) noted "…there is limited value for reducing pavement loading in reducing suspension frequency below 1.5Hz or increasing damping beyond 20%. However, damping greater than 20% could have benefits for heavy vehicle safety performance and for reducing axle hop vibrations under severe conditions of roughness."

The current EC (and Australian) approach to the "damage equivalence" of HVs with RFS at higher masses *vs.* conventional steel HV suspensions have been estimated to overstate the maintenance cost advantage of air-suspended HVs in the UK by 70% (Collop & Cebon, 1997). This view needs to be balanced against the fact that it was made with regard to asphalt fatigue damage, which is not how the great majority of Australian pavements fail. This last issue was addressed by Collop *et al*., (1996) who reported that spatial repeatability was a better predictor of failures in "thin pavements". Given that the great majority of Australian pavements are generally thinner than their European counterparts, this concept needs to be considered in the Australian context (OECD, 1998; Queensland Department of Main Roads, 2007b).

7.5 Potential improvements to VSB 11

VSB 11, p8 (Australia Department of Transport and Regional Services, 2004a) states:

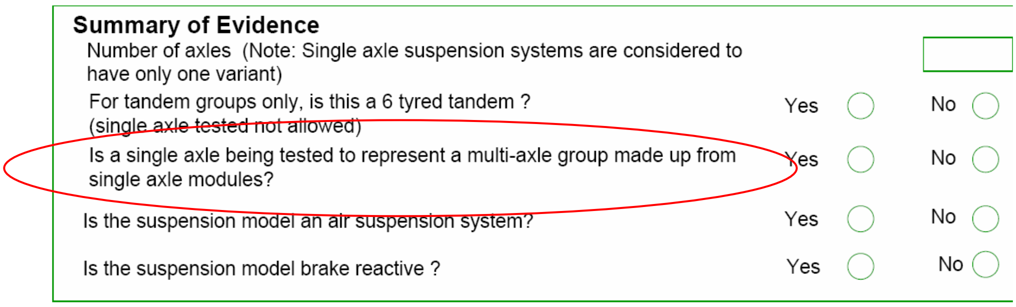
> "A suspension system will be recognised to be road-friendly if it conforms to the following performance and component requirements:
>
> **Performance Requirements**
>
> During free transient low frequency vertical oscillation of the sprung mass above an axle or axle group, the measured frequency and damping with the suspension carrying its maximum legal load must fall within the following limits:
>
> (i)
>
> • The frequency of the sprung mass above the axle or axle group in a free transient vertical oscillation must not be higher than 2.0Hz.
>
> • The mean damping ratio DM must be more than 20% of critical damping (Co) for the suspension in its normal operating condition.
>
> • The damping ratio DR of the suspension with all dampers (if fitted) removed or incapacitated must be not more than 50% of DM.
>
> (ii)
>
> • Static load share between axles in the axle group must be within 5%. (Multiple axle groups only).
>
> (See attached definition of load-sharing suspension system)."

This means, read *verbatim*, that the static load-sharing test is item (ii) under a requirement for free transient vertical disturbance. Put another way, the static load-sharing test to be undertaken during dynamic testing.

Further, although VSB 11 specifies the parameters for RFS in terms of body-chassis parameters, it does not specify any parameters for axle hop, dynamic load-sharing or other potentially road-damaging behaviour of truck suspensions.

As noted in Section 2.4, axle-hop can exert greater forces than body-bounce. Axle hop is not addressed in VSB 11. Undamped HV suspensions without adequate damper operation can exert forces at axle-hop eigenfrequencies at least 500% greater than body bounce frequencies (Sweatman & Addis, 1998). This issue is not addressed under VSB 11.

Section 2.5 outlined how axles are tested under VSB 11. A single axle may be used to represent the behaviours of a number of axles of the same model in a group. The "Suspension evidence" form in Appendix 7 of VSB 11 (Figure 8) provides tacit approval of this practice.


Summary of Evidence

Number of axles (Note: Single axle suspension systems are considered to have only one variant)				
For tandem groups only, is this a 6 tyred tandem ? (single axle tested not allowed)	Yes	◯	No	◯
Is a single axle being tested to represent a multi-axle group made up from single axle modules?	Yes	◯	No	◯
Is the suspension model an air suspension system?	Yes	◯	No	◯
Is the suspension model brake reactive ?	Yes	◯	No	◯

Figure 8. Extract from VSB 11 Appendix 7.

If a single axle is tested as a representative sample of a number of axles in a group, it is difficult to see how the load-sharing ability of that axle group could be determined and certified as such; other than an assumption that an air line between air springs shares the wheel forces.

From the foregoing, it can be judged that VSB 11 needs revision. In addition to addressing the shortcomings listed above, such a review may involve expanding it to encompass in-service suspension testing, particularly low-cost testing.

7.6 "Friendlier" road-friendly suspensions

DLC was used in the analysis to establish the differences between a standard and a modified suspension (Cole & Cebon, 1995). Optimum values for spring constant and damping ratio were determined to minimise road damage (Cole & Cebon, 2007). For their 1995 work, Cole and Cebon:

- chose to find the DLC of the suspensions under test;
- assumed a Gaussian distribution for wheel forces;
- choose the 95th%ile 4th power rule; and
- examined spatial measures in the form of the aggregate 4th power force from earlier work (Cebon, 1987).

These measures were applied to data from an experiment exploring optimisation of HV suspensions for damage by altering spring stiffness and damping ratio on a standard and a modified suspension (Cole & Cebon, 1995). This approach was an evolution of earlier work stating that, whilst the parameters of body bounce and damping ratio (see 2.5) provide information about suspension performance, there was, at the time of developing the tests for "road-friendliness", no relationship between body-bounce frequency or damping ratio and road damage (Cole & Cebon, 1991; Woodroofe, 1995).

Both body-bounce frequency and road damage are influenced by spring rate and damping ratio. It was not until Cole & Cebon (1996; 2007) found optimised HV suspension spring rates and damping ratios that would cause minimum road damage. The optimum spring rate suggested from that work was approximately 1/5th the current industry norm for air-springs. The damping ratio for such a reduced level of road damage was twice that specified by VSB 11 (Cole & Cebon, 1996). Soft spring rates and moderate damping ratios need not to be confined exclusively to air suspension applications (Gyenes *et al.*, 1994; Hahn, 1987b) for improved damage performance from HV suspensions, however. The ranking of "road-friendliness" for different *types* of suspensions also depends on the road damage criteria used (Potter, Cebon, Cole *et al.*, 1996).

7.7 Instrumentation

Axle-hop of a typical heavy vehicle axle is 10 – 15Hz (Cebon, 1999) compared with 2-3Hz for body-bounce (de Pont, 1999). Measuring axle hop involves a higher sampling rate than if body-bounce is to be determined (Chesmond, 1982).

For pavement measurements of dynamic HV wheel-forces, higher sample rates mean WiM slots spaced more closely together and more slots. For on-board measurement, higher sampling frequency will, properly, record more data but use more memory in recording equipment.

The instrumented hub is the most accurate method of measuring dynamic wheel loads. It is also the most complex and expensive (Cebon, 1999).

The most practical method to measure wheel forces on all axles simultaneously is by the use of strain gauges augmented with accelerometers (Cebon, 1999).

For Sweatman’s research (1983) only one hub was instrumented per vehicle due to the cost. That work relied on inferring the other wheel loads as a complement of the measured load which somewhat contradicted earlier work which stated, p5 (Sweatman, 1980): “…instantaneous axle loads will tend to be unequal due to dynamic forces generated over the road profile”

8 Conclusion

This literature review has been undertaken to inform the NTC project developing in-service tests for HV suspensions.

Suspensions are used to smooth the ride for passengers and freight. All vehicles bounce as they operate since completely smooth roads and bridges are impractical and all suspensions are imperfect. The actions of the dynamic forces arising from this bounce, transmitted from vehicle wheels to the ground, causes damage to pavements and bridges over and above the damage done by steady-state wheel forces.

The riskiest portion of the vehicle fleet in terms of road network asset damage is the heavy vehicle (HV) component. The last quarter of the 20th Century was characterised by great efforts into research on the mechanisms for pavement and bridge damage due to HVs. This research has been focussed both on the vehicle; in that suspension types have been assessed for damage, and the network asset; characterised by testing pavement and bridge responses to differing vehicle and suspension types. Despite these large advances, the mechanism for damage to pavements and bridges from the riskiest portion of the vehicle fleet is still not understood entirely. Not letting the perfect be the enemy of the good, robust asset damage models have been formulated and are used widely to estimate damage to the road network asset from HVs. The complementary models for suspension and HV component behaviour have likewise been modelled and reasonable estimates of measures and behaviour may be made. Nonetheless, there is still more work to do.

The transport sector is under continuous pressure from its clients to be more productive and efficient. This was managed in the past with general mass increases. The limit of the network to accommodate those general increases either has been reached or will be reached shortly, depending on the particular network link.

The pressure from the transport sector and its clients does not stop, however, so the heavy transport industry's propensity to innovate has emerged more strongly as an answer to the "pincer movement" of continued client pressure and the slowing of the network ability to absorb more mass. The indicative speed at which vulnerable assets, particularly bridges, can be upgraded currently is approximately 12 per year

(Queensland Department of Main Roads, 2007c) under the timber bridge replacement programme in Southeast Queensland. There are approximately 400 timber bridges in Queensland which are not able to accommodate HML loads (Queensland Department of Main Roads, 2007b). Compare this rate of asset upgrades to the time taken to build an innovative vehicle. It will be some time before the Queensland network can take HML across its entirety; this short summary does not include the damage to pavements as noted above.

Not counting specialised single-trip movements of highly-visible and very high risk vehicles, the mis-match of more HV mass *vs*. network capacity to withstand it was first managed for HVs *en bloc* by the use of restricting particular HVs with higher GVM to certain routes for safety and amenity purposes (e.g. road-trains on remote routes). The next appreciable roll-out of route-specific vehicles was the higher mass limit (HML) schemes around Australia. These required HVs to be equipped with "road-friendly" air suspensions in return for being allowed greater mass as a concession. With the clarity of hindsight, this was probably not managed as well as it could have been given the current understanding of HV air suspensions, particularly the degradation of the asset due to poorly maintained air suspensions. Accordingly, the test regime for air suspensions needs to be revised for both type testing of new HV suspensions and inclusion of in-service testing of HVs that work at HML. Such testing needs to incorporate suspension damper (shock absorber) health tests as these are the critical components in keeping "road-friendly" HV suspensions "friendly".

Considerable savings in pavement rehabilitation could be made by merely enforcing the requirements of the legislation covering "road-friendly" HV suspensions for HML as it stands. After all, correctly operating shock absorbers are a component required for workplace health & safety, roadworthiness and road safety legislation. It should not be an imposition to require the riskiest vehicles on the road to be roadworthy and, in so doing, stay "road-friendly". If the outcome of this is reduced pavement wear so that pavement rehabilitation funds can be directed elsewhere then this becomes a "win-win" scenario for society in general and the transport industry in particular.

An in-service test is being developed by the NTC under a project funded jointly by Queensland, NSW and DoTaRS. This report and the research project *Heavy vehicle* *suspensions – testing and analysis* being undertaken at QUT will inform the NTC project in its work.

Appendix 1 – Derivation of wheel-force measurements

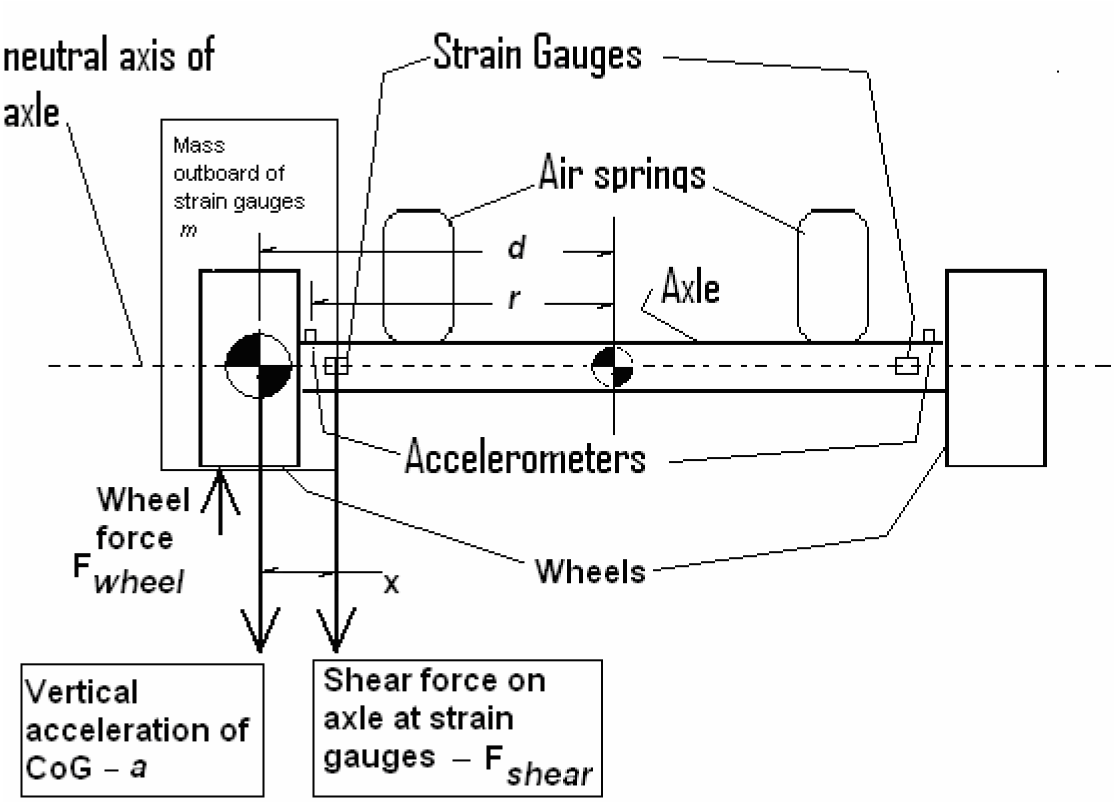


Figure 9. Showing variables used to derive dynamic tyre forces from instrumented HV axle.

Referring to Figure 9 and from the work of previous researchers (Cebon, 1999; de Pont, 1997; LeBlanc *et al*., 1992; Whittemore, 1969; Woodroofe *et al*., 1986), wheel-force may be derived from an instrumented HV axle as shown using the balance of forces on a particular wheel.

$F_{wheel} = F_{shear} + ma$

Where:

a is the acceleration of the mass outboard of the strain gauges;

m is the mass outboard of the strain gauge;

F_{shear} is the shear force on the axle at the strain gauge; and x is the distance from the strain gauge to the effective centroid of the wheel (see Section 4.3.2).

Mounting the accelerometers as close as possible to the hub of the wheel places them, in effect, at the CoG of the mass outboard of the strain gauges. Any small differences between the mounting point and the actual CoG may be neglected if:

- the roll angle is small; and
- the distance from the centre of the axle to the accelerometer approximates to that of the distance from the centre of the axle to the CoG of the mass outboard of the strain gauges;

i.e. when:

$d \underline{\sim} r$ (Cebon, 1999)

and especially when:

$(d - r) \ll d$.

When comparing two different test cases with the same instrumented axle, the error due to $d \neq r$ will be present for both cases and will therefore cancel out (Woodrooffe & LeBlanc, 1987). Further, de Pont noted that large variations in the value of the mass outboard of the strain gauges do not contribute greatly to overall variations in the resultant wheel forces (de Pont, 1997).

Appendix 2 – Extracts from Queensland's BIFA and NSW's BIFA

Extracts of the Queensland and NSW BIFAs are shown below with deletions, for brevity, in [].

NSW (Australia Department of Transport and Regional Services, 2005a):

"71) Both parties agree to support development by the NTC of an in-service suspension test for road friendly suspensions for inclusion in the NHVAS mass management module. The Australian Government will provide supplementary funding to the NTC to assist with accelerating that work and to develop an appropriate enforcement framework.

Pending development of the in-service test the parties also agree to pursue a nationally agreed component replacement programme for road friendly suspensions through an amended NHVAS mass management module developed by the NTC…"

Queensland (Australia Department of Transport and Regional Services, 2005b):

"74) Accordingly, both parties agree to work co-operatively towards ensuring a structured sensible extension of HML vehicle access onto a broader strategic network. It is agreed that further extensions will reflect the following principles:

a) []; and

b) Both parties commit to the development of an enforceable in-service road friendly suspension maintenance regime. This will involve supporting NTC in:

i) development of a proposal for ATC consideration for an in-service suspension test for road-friendly suspensions, with suitable penalties attached for actual damage or non-compliance with the maintenance regime; and

pending 74 b) i above, development of an urgent proposal for ATC consideration to amend current HML arrangements to provide an agreed and enforceable component replacement program.

The Australian Government will provide supplementary funding to NTC for this purpose…"

Appendix 3 – Extracts from RFS test specifications, EC and VSB 11

EC

The European Council Directive 92/7/ECC (1992) specifies the parameters for RFS. These include a damping ratio $\geq$ 0.2 and a natural frequency $\leq$ 2.0Hz. The EC specifies testing methods with substitution allowed as follows:

The loaded vehicle is to be:

Driven at low speed (5km/h +/- 1km/h) over an 80mm step with the transient oscillation to be analysed for frequency and damping after the wheels on the driving axle have left the step;

- Pulled down by the chassis so that the driving axle load is 1.5 times its maximum static value. The vehicle hold down is released suddenly and the subsequent oscillation analysed;
- Pulled up by the chassis so that the sprung mass is lifted by 80mm above the driving axle. The vehicle hold up is dropped suddenly and the subsequent oscillation analysed;
- Lifted so that the sprung mass is raised 80mm and dropped in free-fall with analysis of resultant oscillations; or
- Be subjected to other equivalent procedures as approved by the manufacturer and the satisfaction of the EC.

VSB 11

Test procedure

VSB 11, Section 4 defines the RFS test procedure:

TEST PROCEDURE

To establish by test the damping ratio D, the damping ratio with hydraulic dampers removed, and the frequency F of the suspension, the loaded vehicle should either:

(a) be driven at low speed (5km/hr ± 1 km/hr) over an 80 mm step…. The transient oscillation to be analysed for frequency and damping occurs after the wheels on the driving axle have left the step; or

(b) be pulled down by its chassis so that the driving axle load is 1.5 times its maximum static value. The vehicle hold down is suddenly released and the subsequent oscillation analysed; or

(c) be pulled up by its chassis so that the sprung mass is lifted by 80 mm above the driving axle. The vehicle hold up is suddenly dropped and the subsequent oscillation analysed; or

(d) be subjected to other procedures in so far as it has been proved by the manufacturer, to the satisfaction of the technical department (VSS), that they are equivalent.

Damping, frequency and load-sharing

VSB 11 p8, (Australia Department of Transport and Regional Services, 2004a) states:

> "A suspension system will be recognised to be road-friendly if it conforms to the following performance and component requirements:
>
> **Performance Requirements**
>
> During free transient low frequency vertical oscillation of the sprung mass above an axle or axle group, the measured frequency and damping with the suspension carrying its maximum legal load must fall within the following limits:
>
> (i)
>
> • The frequency of the sprung mass above the axle or axle group in a free transient vertical oscillation must not be higher than 2.0Hz.
>
> • The mean damping ratio DM must be more than 20% of critical damping (Co) for the suspension in its normal operating condition.

• The damping ratio DR of the suspension with all dampers (if fitted) removed or incapacitated must be not more than 50% of DM.

(ii)

• Static load share between axles in the axle group must be within 5%. (Multiple axle groups only).

(See attached definition of load-sharing suspension system)."

VSB 11 p9, (Australia Department of Transport and Regional Services, 2004a):

"**load-sharing suspension system** means an axle group suspension system that:

(a) is built to divide the load between the tyres on the group so that no tyre carries a mass more than 5% greater than the mass it would carry if the load were divided equally…"

Appendix 4 – Derivation of the damping ratio formula: both versions

There is some debate about which formula to use for the derivation of the damping ratio (ζ). Most references usually use the classical derivation from the exponential decay of a 2nd-order system (Meriam & Kraige, 1993):

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}$$

Equation 5

Using Equation 5 and solving for ζ:

=> $$(\sqrt{1-\zeta^2})\delta = 2\pi\zeta$$

=> $$(\sqrt{1-\zeta^2})(\sqrt{1-\zeta^2})\delta^2 = (2\pi)^2\zeta^2$$

=> $$(1-\zeta^2)\delta^2 = (2\pi)^2\zeta^2$$

=> $$\delta^2 - \zeta^2\delta^2 = (2\pi)^2\zeta^2$$

=> $$\delta^2 = (2\pi)^2\zeta^2 + \zeta^2\delta^2$$

=> $$\delta^2 = \zeta^2[(2\pi)^2 + \delta^2]$$

=> $$\zeta^2 = \delta^2 \big/ [(2\pi)^2 + \delta^2]$$

=> $$\sqrt{\zeta^2} = \sqrt{\delta^2 \big/ [(2\pi)^2 + \delta^2]}$$

=> $$\zeta = \delta \Big/ \sqrt{[(2\pi)^2 + \delta^2]}$$

This gives Equation 1 as required.

Some references provide both equations (Meriam & Kraige, 1993):

$$\zeta = \frac{\delta}{\sqrt{\delta^2 + (2\pi)^2}}$$

Equation 1

Appendix 5 – Determining strain gauge output relationship with static wheel load

The following procedure (Woodroofe *et al.*, 1986) has been adopted by researchers (Davis, 2007; de Pont, 1992, 1997; Hoogvelt *et al*., 2004) to determine the relationship of strain gauged axles to wheel forces. Details vary but the concepts are common:

- each wheel of the axle group of interest on the test vehicle was positioned so that its wheel-force could be measured on a certified measurement instrument such as wheel-force scales used for enforcement purposes;
- the chassis of the test vehicle was jacked up so that the wheel-force registered as close to zero as possible on the scales;
- the reading of the strain gauges at the resultant zero wheel-force was set to zero using set potentiometers;
- the reading of the measurement system at the resultant zero wheel-force load was set at zero;
- the chassis was lowered to normal operating mode with the vehicle at tare;
- the wheel-force of each wheel of the axle group being measured on the test vehicle was weighed using the scales, etc. This static wheel-force value was recorded for each wheel at various test loads. The minimum test load being tare and the maximum being full load; the test vehicle being loaded progressively with test weights.
- For each different test load, the reading of the strain gauges at that point yielded a signal that was matched to the calibrated wheel-force via the scales for each wheel.

These readings then provided the offset and slope on the axle strain *vs.* wheel-force graph (Woodroofe *et al.*, 1986) for each axle-end on the axle/s of interest.

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